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U. S. ARMY
TRANSPORTATION RESEARCH COMMAND
FORT EUSTIS, VIRGINIA

TRECOM TECHNICAL REPORT 63-15

FEASIBILITY OF REINFORCED PLASTICS FOR PRIMARY STRUCTURE OF ARMY AIRCRAFT

Task 9R38-01-017-69

Contract DA 44-177-TC-756

March 1963

prepared by:

HAYES INTERNATIONAL CORPORATION
Birmingham, Alabama

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HEADQUARTERS
U. S. ARMY TRANSPORTATION RESEARCH COMMAND
Fort Eustis, Virginia

In this report, the Hayes International Corporation has conducted a literature search, design studies, and laboratory tests in an effort to determine the feasibility of using reinforced plastics as primary structure of Army aircraft.

The report has been reviewed by the Transportation Research Command and is considered to be technically sound. It is published for the exchange of information and stimulation of ideas.

The conclusions and recommendations made by the contractor are considered to be valid by this Command.

FOR THE COMMANDER:

KENNETH B. ABEL
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E. ROUZEE GIVENS
Project Engineer

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FEASIBILITY OF REINFORCED PLASTICS FOR PRIMARY STRUCTURE OF ARMY AIRCRAFT

Hayes International Corporation Report No. 743

Prepared by
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Birmingham, Alabama

for
U. S. ARMY TRANSPORTATION RESEARCH COMMAND
FORT EUSTIS, VIRGINIA

FOREWORD

A study to determine the feasibility of using reinforced plastics for primary structure of Army Aircraft was conducted by Hayes International Corporation under Contract No. DA 44-177-TC-756 for the U. S. Army Transportation Research Command, Fort Eustis, Virginia. The contract was initiated in July 1961 and was concluded in January 1963.

The program was conducted under the direction of Mr. J. N. Daniel, Chief of Systems and Equipment Division; Mr. J. E. Forehand, Chief of Aircraft Components and Accessories Branch; and Mr. E. R. Givens, Project Engineer; Aviation Directorate, USATRECOM.

Principal Hayes engineers were L. R. Anderson, Project Engineer; C. L. Anker, R. S. Brown, A. E. Dietz, V. E. Morrow and R. B. Wysor - Analysis; C. B. Reymann - Materials and Processes; P. T. Howse - Test; J. F. Davenport, R. A. Holder and A. M. Smallwood - Design. The program was under the technical direction of B. A. Reymann.

Government and industry sources of information are credited in the text or are noted in the list of references. Special recognition is given to Summit Industries for the fabrication of test specimens, to Hercules Powder Co. for supplying technical data on filament winding, and to Minnesota Mining and Manufacturing Co. and Bloomingdale Rubber Co. for adhesive bonding and testing.

No specification for plastic materials are included in this report; trade names for plastic materials have been included for the sole purpose of identification.

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SUMMARY

Recent advances in the technology of reinforced plastics have produced composite materials which have strength properties equal to those of heat-treated steel with weights approximately the same as magnesium. These materials have been used quite extensively in nonstructural parts for aircraft. Recently their use in secondary structural applications and some primary structure has steadily increased. The objective of this program was to determine the feasibility of using reinforced plastics in highly stressed Army aircraft structures and components by design studies and the fabrication and testing of reinforced plastic specimens. This document is the final report of the investigation. It contains the results of the design studies, the results of all tests, conclusions, and recommendations.

Requirements for the pertinent structures and components were established to insure compliance with applicable specifications, criteria, and Army directives. Design studies of various components were then accomplished resulting in preliminary reinforced plastic configurations. These included fuselage, wing, empennage, landing gear struts, power transmission shafts, transmission housings, and fuel tanks. Available data on work that has been accomplished by other organizations on the use of reinforced plastics in rotor and propeller blades were summarized.

Fiberglass offers the higher mechanical properties of the several reinforcing materials. Therefore, it is used exclusively for this study. It was concluded that fiberglass reinforced plastics are feasible materials for use as primary structure for Army aircraft. Specific advantages can ge gained by their use in helicopter tail booms, landing gear shock absorbing struts, rotor blades and small control surfaces. Other structural components indicate feasibility but require further investigation and evaluation.

The main limitation to the use of fiberglass reinforced plastics for structure is their low modulus of elasticity. However, in some applications, such as landing gear struts, a low modulus of elasticity is an advantage rather than a disadvantage. This study has indicated that presently available materials are feasible for some types of primary structure. When the special high modulus glass fibers currently under development are fully developed, it is reasonable to believe that glass reinforced plastics will become a highly feasible and competitive material for use in all primary structure.

It is recommended that the study program be continued to include the design, fabrication, and test of full scale components for specific applications.

CONCLUSIONS

Reinforced plastics are considered feasible materials for use in primary structure of Army aircraft and offer advantages over conventional metal structures for certain components and requirements. These materials are specifically feasible for the following structures and components and effort leading to the development of hardware is justified.

- Helicopter Tail Booms
- Aft Body of Light Fixed Wing Aircraft
- Helicopter Skid Type Landing Gear
- Fixed Cantilevered Landing Gear Struts
- Helicopter Control Surfaces
- Fuel Tanks

Feasibility of the following items is indicated, but further investigation and evaluation is required.

- Light Fixed Wing Aircraft Wings
- Light Fixed Wing Aircraft Empennages
- Transmission Housings

Feasibility of rotor and propeller blades is indicated by the work of others, but has not been evaluated.

The use of reinforced plastics in helicopter tail booms, control surfaces and similar components results in less weight, better aerodynamic efficiency, better appearance, radar transparency, and durable structure with good fatigue characteristics at costs that would be comparable to or lower than metal components. Wing and empennage structure indicates similar advantage, but the evidence is not conclusive.

Reinforced plastics are excellent energy absorbers. Their use for landing gear shock absorbing components will reduce the landing load factor for normal rates of descent resulting in less wear and tear on aircraft structure and equipment, and greater comfort for the occupants. Conventional "yielding" metal landing gears on helicopters require replacement after "hard" landings. A reinforced plastic shock absorbing strut could react loads from a "hard" landing without failure.

The irregular contours characteristic of most aircraft fuel tanks would be readily adaptable to the advantageous use of reinforced plastics. Such a tank would be lighter than a metal tank and would have superior corrosion resistance and potentially less fatigue problems for given vibration conditions. A reinforced plastic tank would be adaptable for quantity production at a cost that would be comparable to or lower than that for metal tanks.

The use of reinforced plastic in transmission housings may result in some advantages but considerably more study is required. This application is somewhat questionable. Reinforced plastics for power transmission shafts are not warranted unless the resistance to environmental conditions or radar transparency properties are required.

The noise in an aircraft may be reduced by reinforced plastic structure, but it is doubtful if the reduction would be significant. Some advantage can possibly be realized. Integrally molded components resulting in relatively large single-piece construction will reduce the direct air transmission of noise. The acoustic properties require further investigation

The materials and manufacturing process for a specific component must be chosen for the specific requirements, environment, configuration, quantity, etc. It is believed that more consistent results in the fabrication of most components can be realized with epoxy resin than with polyester resin.

The curing cycle for all materials, especially for sandwich construction, can be quite critical. Unless the fabricator has had experience with the materials and cure cycle, some developmental work will be required to insure optimum results and compatibility of materials.

RECOMMENDATIONS

It is recommended that the structures and components conclusively indicating feasibility be considered for early development in the following order.

Helicopter Skid Type Landing Gear
Helicopter Tail Boom
Helicopter Control Surface
Fixed Cantilevered Landing Gear Strut
Fuel Tank

A comprehensive study of wings, body, and empennage structure required an effort greater than was feasible in this program. It is recommended that further study be accomplished on these components. Additional study is also required for transmission housings.

It is recommended that development of the reinforced plastic landing gear strut for a specific fixed-wing aircraft and/or a helicopter be initiated as expeditiously as practical, and to include the following:

1. Additional analytical studies aimed at sandwich type construction as well as solid laminates and new materials with higher strength to develop a design for a landing gear strut for a particular aircraft.

2. Accumulate data on design and service experience with gear of this type now in use.

3. Accomplish strength and fatigue testing of specimens of beams using construction methods decided upon through analytical studies.

4. Fabricate full scale components and accomplish strength and fatigue tests.

5. Install reinforced plastic landing gear on aircraft and accomplish drop, flight, and service tests.

A similar program is recommended for the development of a helicopter tail boom.

There is some indication that aircraft noise can be reduced by the use of reinforced plastic structure. In order to evaluate the noise reduction characteristics further, a study program is recommended. This program should include the following:

1. Tests to obtain quantitative data on damping of plastics.

2. Determination of sources of aircraft noise and transmission paths in selected aircraft.

3. Preparation of a preliminary reinforced plastic design of the aircraft body structure and an analytical evaluation of noise transmission characteristics of both designs.

4. If it is concluded that the reinforced plastic design has possibilities of reducing the noise level, fabricate a full scale component and test.

INTRODUCTION

The purpose of this contract was to conduct a research study and test program to determine the feasibility of using reinforced plastics as primary structural materials in Army aircraft. The program was divided into two phases, the first being essentially the design study and the second the test phase. Study was directed at structure and component design requirements in current and future Army aircraft with a view toward replacing existing manufacturing techniques with reinforced plastics in those areas in which a definite advantage seems apparent.

Components of primary structure are those parts of the aircraft in which a failure would result in the probable loss of the aircraft. It has been established, by virtue of many successful structural applications, that reinforced plastics are acceptable structural materials. Relatively few applications have been made in the field of aircraft primary structure; however, the uses have increased extensively in recent years. Several significant structural applications in newer high speed jet transport aircraft are good examples of their recent acceptance as a structural material.

The conduct of the subject program was based on a direct approach to the determination of feasible Army aircraft reinforced plastic applications. Feasibility, in this case, is restricted to those applications where reinforced plastics are advantageous compared to conventional materials. The approach was further based on permitting early achievement of feasible reinforced plastic hardware, where feasibility is indicated.

The program effort was therefore concentrated on those significant Army aircraft structures and components which appear to have the greatest degree of potential feasibility. The following catagories were selected for study:

1. Fuselage
2. Wing
3. Empennage
4. Rotor and Propeller Blades
5. Landing Gears
6. Fuel Tanks
7. Drive Shafting
8. Transmission Housings

Within these categories, those configurations and problem zreas most pertinent to future Army applications were given priority.

Requirements for the pertinent structures and components were established to insure compliance with applicable specifications, criteria and Army directives. Reinforced plastic designs are then developed in accordance with these requirements. The various alternatives in reinforced plastic design and fabrication are explored for the most promising approaches. These configurations were evaluated with respect to each other and with respect to conventional materials.

This report summarizes all work accomplished. It includes design studies, test results, conclusions and recommendations.

When this program was initiated it was anticipated that a substantial amount of data and results of similar studies evaluating reinforced plastics versus other materials accomplished by other sources would be made available to this contractor to aid in the investigation. Many members of industry and Government agencies were contacted requesting such data. Very little pertinent information was obtained in this manner. Industry in general considers that its data are proprietary and therefore would not make them available. Some indicated a desire to cooperate but did not have their studies in a published form that could be used.

A substantial number of reports on basic materials research and substantiating data for MIL-HDBK-17 were obtained from Government sources such as Forest Products Laboratory and The Armed Services Technical Information Agency. There is evidence that many Government-sponsored projects have been accomplished relative to the use of reinforced plastics, the results of which would be beneficial to a study of this type; however, there is no straightforward way of finding and obtaining the documents that report the results.

REQUIREMENTS

The definition of the requirements for the various structures and components to be studied in this program is essential for two primary reasons:

1. To insure that the reinforced plastic designs generated are in conformance to applicable criteria.

2. To provide a true basis for the evaluation of reinforced plastics feasibility in the applications studied.

In order to establish requirements, pertinent specifications, manuals and related publications were reviewed for applicable criteria. This information was supplemented by projected future requirements for Army aircraft based on the available data and this Contractor's experience and judgment. Considerations relative to the Army aircraft mission and service environment were taken into account in the design studies. The potential of the various categories and types of aircraft were considered in establishing the components and priority for study.

The general requirements for all components investigated in this program were in accordance with applicable Army specifications and procedures. Since most Army aircraft were procured to FAA or Air Force specifications, the following general publications were used as guides for the overall aircraft design criteria and structural load requirements:

1. ARDC Manual 80-1, Handbook of Instructions for Aircraft Designers (Reference 7).

2. Civil Aeronautics Manual 4, Airplane Airworthiness (Reference 21).

3. Civil Aeronautics Manual 6, Rotorcraft Airworthiness (Reference 22).

4. MIL-S-8698, Structural Design Requirements, Helicopters (Reference 50).

5. MIL-H-8501, Helicopter Flying Qualities, Requirements for (Reference 49).

6. MIL-S-8785, Flying Qualities of Piloted Airplanes (Reference 51).

7. MIL-S-5700 through MIL-S-5706, Structural Criteria, Piloted Airplanes (References 41 - 46).

The specific requirements for each design are presented in the discussions of the various studies.

die surface. Bag-molded surfaces may be rough. Wrinkles, resin ridges and fabric laps occur on bag-molded sides of laminates and could require subsequent smoothing operations.

The desired color can be obtained by the use of surface paints, gel coats or color pigment added to the laminating resin. The use of gel coats for structural components is not recommended. It gives a low strength resin rich surface.

The use of prototypes in the development of a component is desirable wherever feasible. It allows the evaluation of a component under design loads, environment, and simulated service life conditions. Much can be learned from tests of a prototype that will make the final designs more valuable. Variables that are peculiar to the specific design and method of fabrication can be accounted for in the design to minimize any adverse effect. Where matched metal die molding is to be used, the design should be thoroughly and completely worked out before the molds are made. Changes can be very costly and time consuming.

The strength of a glass reinforced part can be partially dictated by the molding procedure because the process and technique used can control the glass content and quality of the final part.

The choice of molding procedure is a basic consideration in the design of a part. The molding process for a given part is chosen by giving proper consideration to the following:

1. Strength requirements
2. Size of part
3. Shape of part
4. Permissible tooling costs
5. Permissible costs per part
6. Appearance requirements
7. Delivery time
8. Total number of parts to be made
9. Dimensional tolerance requirements

The design of primary structure using reinforced plastics is attractive from a number of viewpoints. High strength/weight ratios for appropriate orientation of load and reinforcement; the ability to build up local areas readily for stiffening and load concentration; the great variety of reinforcements and resins available for selection - these are only a few reasons the engineer is furnished great freedom of design and can achieve weight, cost and fabrication efficiencies not realized with conventional materials.

3. Consider the special characteristics of reinforced plastics and their differences from conventional metal materials.

4. Analyze the structure using appropriate formulae.

5. Consider the inherent stability of glass reinforced plastics.

6. Remember that glass reinforced plastics cannot be sprung.

7. Use color for appearance and permanent finish.

8. Work closely with the mold maker and the molder.

9. Fabricate and test a prototype.

One of the greatest advantages of glass reinforced plastics can be gained from the successful integration of many parts into one unit. This can result in economy because there is less part design, tooling, fabrication, part handling, fastening, inspection, and weight.

The molding processes and techniques for fiberglass reinforced plastics permit a wide flexibility in the shape and form of the final part. Metal structures frequently are overdesigned because of standard gauge materials. In fiberglass reinforced plastics, there is practically no limit to the "tailoring" of thickness distribution and special shape that can be obtained for the most efficient structure. Curved structures provide additional strength and regidity and can be obtained with comparative ease.

Methods of analysis used in the design of metal structures are in general applicable to reinforced plastics provided due allowance is made for the difference in material properties and consideration is given to the fact that the usual fiberglass reinforced plastic cloth structure is an anisotropic material. It should be remembered that a stress analysis is not necessarily satisfactory proof that the structure is adequate. There are many intangibles associated with the fabrication of reinforced plastics that indicate development by testing even more than with metal structures.

Glass reinforced plastic material has great inherent dimensional stability. Properly and completely cured moldings of this material will not yield in the sense that most metals yield. Distorted parts will return to the original shape when the load that cuases distortion is released. Parts fabricated with sheet metal can sometimes be reshaped slightly after they have been formed to fit adjacent parts. This is not true for reinforced plastic parts and therefore these parts must be formed with a high degree of accuracy. The molding technique must provide a good fit to prevent built-in stresses in the final part.

Molded finishes vary from mirror smooth to rough. Surfaces molded against die surfaces are generally the smoothest and are direct reflections of the

Fiberglass, the most generally satisfactory reinforcing material for plastics, produces a family of materials with a wide variety of cost and performance characteristics offering the most advantages for highly stressed components. Fiberglass reinforcements are supplied as continuous strands, fabrics, mats, chopped strands and other forms. Many types of resins are used to give a wide range of mechanical, thermal, and chemical properties. Polyester resins are the most common because of their low cost and ease of fabrication. Epoxy resins are most often selected where high mechanical properties are required. Other resins such as phenolics, silicones and acrylics are used where their special characteristics are desired. The combinations of glass and resin can be controlled by the designer to meet a wide range of performance and cost specifications.

Many processes are available to produce the desired combination of design performance and economy. Each process has its own usefulness for combining different kinds and amounts of glass and resin. Processes vary in ability to utilize different arrangements of glass, amounts of glass, and different resins. A given combination of raw materials, required to meet performance criteria in a given application, narrows the choice of processes to those which can successfully and economically form the raw material into a completed part.

Economical cost and performance result from good design based on judicious selection of both raw materials and processes. Proper materials must be combined in a process so that potential performance is realized at economical cost. Design of the part must take advantage of the material and turn potential limitations into advantages.

The many choices of material and processes put the task of determining the feasibility of using reinforced plastics for a specific application on the designer. He must have a thorough knowledge of the relative merits of all materials and processes. It is not the intent to present complete information on reinforced plastics in this document. A number of text books, Government documents, and manufacturers data books are available that adequately describe the various fabrication processes and tabulate data on materials.

The advantages and limitations of glass reinforced plastics are unique and different from other materials. When designing with these materials, advantage must be taken of their particular characteristics. The design must not be dictated by the performance and characteristics of the more conventional materials.

The general considerations for the design of reinforced plastics can be summarized as follows:

1. Integrate design to minimum number of moldings or parts.

2. Use curves and "streamlined" shapes as required.

TECHNICAL CONSIDERATIONS

Various technical considerations relative to the design and analysis of reinforced plastic structures are applicable to all design studies accomplished in this program. Brief discussions of design, strength, and aerodynamic and dynamic considerations are included for clarity of the report and to avoid repetition of these considerations throughout the design studies.

DESIGN

Accurate analytical determination of the distribution of stresses in aircraft structures is a complex undertaking even with materials whose elastic properties are essentially the same in all directions. With a material such as glass fiber reinforced plastics, in which both the reinforcement and resin properties vary widely, depending upon reinforcing fiber orientation, proportion of fiber to plastic, type of fiber and resin, etc., the problem becomes even more difficult. These difficulties, however, have their compensative advantages. The great variety and versatility of reinforced plastics give the designer a wide choice for maximum satisfaction of the design requirements. In order for this advantage to be fully realized, however, the necessary design data must be available, and the designer must apply the basic principles of good plastics design.

For optimum plastics design, the material characteristics must be used to advantage. Integral design/fabrication is usually feasible and desirable in that the number of pieces to be fabricated, handled, and assembled is reduced. Adhesive bonding is a similar advantage in eliminating fasteners and fabrication operations.

For some time, fiberglass reinforced plastics have been accepted as satisfactory materials for many aircraft components. Their use has mainly been confined to secondary or nonstructural applications, or items requiring the special characteristics of these materials. The high strength to weight ratio, resistance to corrosion and weathering, and ease of fabrication give reinforced plastics an advantage over many other more "conventional" materials for many structural applications. In recent years, their consideration as a primary structural material for aircraft has been rapidly increasing. The reluctance to accept them for use in primary structure is believed to be primarily due to the lack of good reliable design information, the many variables that affect the characteristics of the end product, and the lack of the necessary methods of reliable quality control. This leads to much controversy over whether these materials are satisfactory for aircraft structural components, and it is a generally accepted conclusion that any discussion of their merits include the words "it all depends".

applications of reinforced plastics in the B-58. It also contains a reinforced plastic radome.

H-21 - Rotor blades of fiberglass reinforced plastic were designed and fabricated for the H-21 helicopter. These blades failed on ground test.

H-43B - Glass reinforced plastic rotor blades have been successfully produced for this aircraft. Kaman is at present in production of these blades and is awaiting approval for use on service aircraft. The vertical fins are also fabricated from fiberglass.

Marvel - The Mississippi State University has a research program in which it is planned to build an all-plastic aircraft. The prime consideration in this application is to provide aerodynamic smoothness for improved performance.

MF1-10 - The Swedish firm of A. B. Malmo Flygindustri has built a STOL type aircraft which uses a reinforced plastic landing gear strut. It reportedly provides better damping, lighter weight, and improved shock absorbing features than a metal strut.

F8U-2NE - The wing tips, engine harness cover, and a fuel cell on this Navy jet fighter manufactured by Chance Vought Corporation are significant examples of structural uses of reinforced plastics in newer aircraft. Phenolic resin and glass cloth are used for the engine harness cover. Polyester resins and glass cloth are used for the wing tip and fuel cell.

THE STATE OF THE ART

Reinforced plastics have been widely adapted to a great variety of products. They have been used to a considerable extent on aircraft and missiles, but despite their apparent feasibility as structural parts, they have been employed very little for primary structure production applications. Experimental or small quantity applications do not have the significance of production uses since feasibility, in the sense of the subject study, is not necessarily indicated.

For wide acceptance as a material for primary structure, it is necessary that the state of the art of reinforced plastics progress to the point where raw materials fabrication process controls and material properties are well established. The design engineer can then design around material properties which are documented by ample test data and not have to introduce raw material and process control variables into his considerations.

The progression of the technology of reinforced plastics is actually hampered by some of the same factors which give these materials structural advantages. For example, reinforced plastics can take advantage of directional properties to design a more efficient structure, compared to metals, in certain applications; yet, the wide variation in directional properties is one of the additional considerations imposed on the design engineer. The same analogy can be applied to variables such as the resins, reinforcements, cure cycles, fabrication techniques, and tooling methods.

It is apparent that the major handicap to feasible reinforced plastic structural applications is a lack of available general knowledge. Even basic engineering design information is extremely sparse from a structural standpoint. This does not mean that there is a lack of data. There is, in fact, such a great bulk of uncoordinated data that it magnifies the engineer's problems. As a result, the designer can determine one or more potential solutions to his problem with relative ease. But the optimization of his solution involves considerable difficulty.

There is essentially no information available on the subject of reinforced plastics feasibility for structural applications. What information exists in industry is considered proprietary and is believed to be generally restricted in scope.

The situation with regard to fabrication knowledge is somewhat better than design knowledge. In this case, secondary structural or even nonstructural experience may be pertinent to primary structural applications; however, in some cases, lack of knowledge of proprietary methods will inhibit the evaluation of problem solutions.

The design engineer will always be faced by volumes of data which he can distinguish as qualification test data or quality control test data. For example, the most popular type of test result reported in all specifications and manufacturers' literature is the flexure test. Now the flexural strength does not fit into the needs of the design engineer for use in his structural analysis formulae. Therefore, it is important to keep in mind the gap between the great volume of data available and the relatively small amount of practical use in conventional design analysis. For reinforced plastics to progress in use for structural application, it will be necessary to conduct suitable tests on special test panels to develop the required design analysis data to support widespread use of this material.

The use of reinforced plastics for primary structural applications in aircraft has developed more slowly. At present there are several outstanding uses of reinforced plastics in aircraft primary structure. Following are some of these applications:

Boeing 707 - The 707 jet airliner has approximately 720 reinforced plastic items. Most of them are nonstructural. A section of the leading edge extension is considered an example of a primary structural application. Several highly loaded items of secondary structure include the nose radome and the large tail cone.

Convair 880 - The upper part of the vertical stabilizer is used as an antenna and must be isolated electrically from the other structure. A splice section of fiberglass reinforced plastic used as a separator must carry all airloads from the upper section and therefore is considered primary structure.

DC-8 - The Douglas DC-8 uses the upper section of the vertical stabilizer as an antenna. A reinforced plastic separator is used in the same manner as on the Convair 880. Another structural application is a 16-foot-long dorsal fin that is molded in one place.

F-27 - The wing trailing edge and the leading edges of all movable surfaces are fiberglass reinforced plastic.

PA-29 - The Piper Aircraft Corporation is now building an all-plastic aircraft for the low-cost private airplane market. It reportedly is made of a paper honeycomb sandwich with 1/32-inch reinforced plastic skins. The fuselage and wings are made in two halves and then joined. The wing contains no ribs or stringers and reportedly has better fatigue life than an equivalent metal wing. Information from Piper was not available.

B-58 - The Convair-built Hustler bomber contains wing sandwich panels with reinforced plastic facings. It also has a large reinforced plastic sandwich panel in a section of the fuselage. There are many nonstructural

applications of reinforced plastics in the B-58. It also contains a reinforced plastic radome.

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For optimum plastics design, the material characteristics must be used to advantage. Integral design/fabrication is usually feasible and desirable in that the number of pieces to be fabricated, handled, and assembled is reduced. Adhesive bonding is a similar advantage in eliminating fasteners and fabrication operations.

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1. Integrate design to minimum number of moldings or parts.

2. Use curves and "streamlined" shapes as required.

3. Consider the special characteristics of reinforced plastics and their differences from conventional metal materials.

4. Analyze the structure using appropriate formulae.

5. Consider the inherent stability of glass reinforced plastics.

6. Remember that glass reinforced plastics cannot be sprung.

7. Use color for appearance and permanent finish.

8. Work closely with the mold maker and the molder.

9. Fabricate and test a prototype.

One of the greatest advantages of glass reinforced plastics can be gained from the successful integration of many parts into one unit. This can result in economy because there is less part design, tooling, fabrication, part handling, fastening, inspection, and weight.

The molding processes and techniques for fiberglass reinforced plastics permit a wide flexibility in the shape and form of the final part. Metal structures frequently are overdesigned because of standard gauge materials. In fiberglass reinforced plastics, there is practically no limit to the "tailoring" of thickness distribution and special shape that can be obtained for the most efficient structure. Curved structures provide additional strength and regidity and can be obtained with comparative ease.

Methods of analysis used in the design of metal structures are in general applicable to reinforced plastics provided due allowance is made for the difference in material properties and consideration is given to the fact that the usual fiberglass reinforced plastic cloth structure is an anisotropic material. It should be remembered that a stress analysis is not necessarily satisfactory proof that the structure is adequate. There are many intangibles associated with the fabrication of reinforced plastics that indicate development by testing even more than with metal structures.

Glass reinforced plastic material has great inherent dimensional stability. Properly and completely cured moldings of this material will not yield in the sense that most metals yield. Distorted parts will return to the original shape when the load that cuases distortion is released. Parts fabricated with sheet metal can sometimes be reshaped slightly after they have been formed to fit adjacent parts. This is not true for reinforced plastic parts and therefore these parts must be formed with a high degree of accuracy. The molding technique must provide a good fit to prevent built-in stresses in the final part.

Molded finishes vary from mirror smooth to rough. Surfaces molded against die surfaces are generally the smoothest and are direct reflections of the

die surface. Bag-molded surfaces may be rough. Wrinkles, resin ridges and fabric laps occur on bag-molded sides of laminates and could require subsequent smoothing operations.

The desired color can be obtained by the use of surface paints, gel coats or color pigment added to the laminating resin. The use of gel coats for structural components is not recommended. It gives a low strength resin rich surface.

The use of prototypes in the development of a component is desirable wherever feasible. It allows the evaluation of a component under design loads, environment, and simulated service life conditions. Much can be learned from tests of a prototype that will make the final designs more valuable. Variables that are peculiar to the specific design and method of fabrication can be accounted for in the design to minimize any adverse effect. Where matched metal die molding is to be used, the design should be thoroughly and completely worked out before the molds are made. Changes can be very costly and time consuming.

The strength of a glass reinforced part can be partially dictated by the molding procedure because the process and technique used can control the glass content and quality of the final part.

The choice of molding procedure is a basic consideration in the design of a part. The molding process for a given part is chosen by giving proper consideration to the following:

1. Strength requirements
2. Size of part
3. Shape of part
4. Permissible tooling costs
5. Permissible costs per part
6. Appearance requirements
7. Delivery time
8. Total number of parts to be made
9. Dimensional tolerance requirements

The design of primary structure using reinforced plastics is attractive from a number of viewpoints. High strength/weight ratios for appropriate orientation of load and reinforcement; the ability to build up local areas readily for stiffening and load concentration; the great variety of reinforcements and resins available for selection - these are only a few reasons the engineer is furnished great freedom of design and can achieve weight, cost and fabrication efficiencies not realized with conventional materials.

STRENGTH

Methods of analysis used to design metal structure are in general applicable to reinforced plastic structures. Strength properties of glass reinforced laminates may vary considerably, and differences of several hundred percent may be found in some properties, depending upon the type of reinforcement and upon the characteristics of the individual reinforcement within a type. Fabrics may be woven such that they have different strength properties in the two directions parallel and perpendicular to the warp direction. Further versatility in materials is possible by cross-laminating or by combining various fabrics in a single parallel laminate. Thus, a wide range of properties is available to the designer, enabling him to fit his materials to the particular requirements of his application. Along with this greater versatility, there is a greater responsibility for the designer to apply those properties toward realization of a more optimum structure.

Aside from the consideration of the basic strength qualities of the various materials, the designer must recognize and allow for the effect of environment and loading conditions on these properties. Environmental conditions that affect strength include temperature, humidity, weathering (including erosion and corrosion), fungus and chemical action. The different loading conditions that may or may not affect strength include duration of loading, rate of loading and frequency of loading.

Finally, due consideration must be given the manufacturing processes and quality control techniques and their effect on the consistency of the mechanical properties of the finished product.

The variation of strength properties of glass fiber reinforced plastics with change in temperature is dependent on the laminating resin and the glass fiber used. Generally, there is an increase in strength with a decrease in temperature below normal and a decrease in strength with increasing temperature. However, in the range of atmospheric temperatures involved, there is only a minor effect on strength for most glass reinforced plastic. In areas where higher temperatures are involved, for example, in the area of a turbine engine exhaust, special attention must be given to this problem.

When exposed to free water or high humidity, glass fabric laminates absorb moisture. This moisture absorption results in an appreciable loss in strength. This reduction is apparently a function of moisture content at the time of loading rather than a permanent deterioration of the material. If the laminate is "dried out" after exposure, it regains its strength. All design allowables used in this study have been based on wet strength. The use of wet strength values is considered to be unnecessarily conservative for most aircraft applications. The conditions under which the wet strength is determined are considered to be unrealistic when related to actual aircraft environment and condition of material when subjected to the design loads. In addition, the laminates can be protected with resin coatings to prevent the absorption of water.

Atmospheric exposure affects the strength properties of glass fiber reinforced plastics, the magnitude of the effect depending primarily on the type of resin and atmospheric conditions. The greater portion of the reduction in strength results from surface erosion. The loss in strength due to weathering for laminates utilizing polyester resins is quite appreciable; however, by painting or other surface treatment, this loss can be reduced appreciably. The effect of exposure on laminates using epoxy resin is negligible.

Mold organisms have been observed to grow on glass-fabric laminates; however, there is little indication that this growth had any effect on properties.

Glass reinforced plastics are quite resistant to attack by most common chemicals. Aviation fluids, fuel, oil, etc., have no appreciable effect on strength properties. Reinforced plastics present somewhat higher creep values than do the common structural metals at comparable temperatures. In nearly all aircraft structural applications the structure is designed to large magnitude, short duration loads whereas the steady state loads are only a small fraction of the design loads. For this reason, the effect of duration of loading on strength as applied to reinforced plastics is not critical in most cases.

The rather limited data available indicates that the rate of loading has little effect on strength at the higher rates. At lower rates, the strength is reduced due to creep rupture.

Consideration of cyclic loading is important in strength evaluation since it is one of two requisite conditions for fatigue, the other being a certain minimum stress level known as the endurance limit. Of course the greater the stress, the fewer the cycles required to result in failure. The stress producing failure is the maximum stress within the member. The maximum stress may be several times the stress predicted by elementary stress theories because of stress concentration. This stress concentration occurs when the application of loading is localized or when the stress pattern is disturbed by eccentricities or discontinuities in the structure. Discontinuities may arise from such things as holes for attachments, necessary changes in section, or from imperfections in the structural material.

Considerable test work has been accomplished by the Government and industry to determine the effects of stress concentrations on both static strength and fatigue strength of reinforced plastics. The results of much of these data are summarized in MIL-HDBK-17. Generally, these data show a rather wide variation in the effect of stress concentrations depending on the following most significant factors:

1. Type of resin employed
2. Resin content
3. Type of reinforcement
4. Fiber finish in the case of glass

5. Fiber orientation
6. Temperature
7. Types of stress concentration
8. Magnitude and characteristics of the stress

As with metals, stress raisers such as holes, cutouts, notches and fillets greatly affect the strength capability of structure fabricated of reinforced plastics. There are, however, definite differences in the behavior of the two materials under such internal stress distribution. Most metals when tested for fatigue develop cracks originating at a point of stress concentration. In reinforced plastics, stress concentrations induce premature failures not only after numerous load cycles, but also during application of a steady load.

This notch sensitivity of reinforced plastic laminate is directly related to the stress-strain behavior of the material. A contrast of the tensile stress-strain curves of 181 glass fabric-polyester laminate with a high strength aluminum alloy is presented in Figure 1. It is apparent that in the case of the aluminum, the rate of straining greatly increases after the yield point is reached. This fact allows the stress concentration to redistribute in adjacent areas, thereby rounding off the theoretical peak stresses. The stress-strain curve for glass reinforced plastic laminates is essentially linear up to the point of failure without the greater plastic flow. Due to this difference, the redistribution of the stress concentration is considerably less than in the case with the metal, resulting in relatively higher concentration factors.

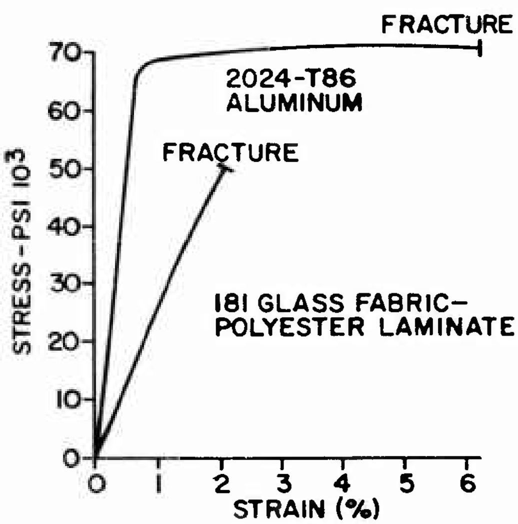


FIGURE 1. STRESS STRAIN CURVES-ALUMINUM ALLOY AND FIBERGLASS LAMINATES

Another interesting fact appears to be inherent with plastic laminate structure. Tests run by the Martin Company, Baltimore, indicate that the greater the number of holes in a given area, the lower the safe stress level. It is reasoned that the larger number of holes increases the probability of early crack formation and propagation. This fact is recognized in industry by the use of special diamond drills to insure sharp, clean holes with minimum delaminating and fraying.

Comparing the more common structural types of glass reinforced plastics to the aluminum alloys employed in aircraft structures, the fatigue strength of unnotched specimens is generally equivalent when compared to ultimate strength. However, as opposed to static conditions, the reinforced plastics are somewhat less notch-sensitive in fatigue.

Stress concentration cannot be avoided entirely in a practical structure; however, reinforced plastics have an advantage over metals in reducing the number and severity of these concentrations. They can be molded to shapes that provide smooth transition of load paths and the number of parts is reduced, thereby reducing the number of joints that are a source of stress concentrations. Perhaps the biggest disadvantage with plastics is the difficulty of maintaining really close control throughout fabrication and the lack of simple, nondestructive inspection techniques.

A general requirement in the design of metal aircraft structures is a positive margin of safety when comparing maximum design loads to the yield stress of metal and a 1.5 safety factor when comparing these loads to the ultimate or breaking strength of the material. Since glass reinforced plastics do not have a yield point as such, this same requirement applied to plastics results in a 1.5 safety factor without yield.

Because of the lesser amount of experience with plastic structures, there is a tendency to apply an additional factor of safety when using this material, presumably to allow for the following factors:

1. Incorrect assumptions on which the analysis and computations are based
2. Effects of temperature changes
3. Effects of repeated stresses
4. Effects of dynamic loads and vibrations
5. Effects of stress raising discontinuities
6. Effects of environment
7. Service conditions
8. Possible increase of loads through future "growth"
9. Variables of workmanship
10. Dependability of quality control
11. Material variations

It is proposed that unless this factor is exorbitant, it will be insufficient to cover all of the variations in particular cases while in many instances it will invoke undue penalties. It is believed that there is sufficient data to permit reasonable allowances for these variations in particular applications, resulting not only in better overall strength but also in greater economy.

The allowable design stresses for materials used in these studies were obtained from MIL-HDBK-17, Plastics for Flight Vehicles, Reference 38, and ANC-23, Sandwich Construction for Aircraft, Reference 6. For materials not included in these two documents, allowable stresses were determined from manufacturers' data, test reports, or other suitable sources.

DYNAMICS

The dynamic behavior of a structure in a given environment can be described in terms of mass, stiffness, and the degree of damping involved. In many cases there is a rather complicated relationship between the properties of the structural configuration and the environmental conditions; however, variation of any of the material properties has the same basic effect under any conditions. Dynamic considerations involve a wide range of environmental conditions including such things as response to impulsive loading, response to periodic or random type loading varying in frequency from relatively low values to sonic frequencies, and self-induced or sustained oscillations such as flutter. Finally, the response of a structure to these dynamic conditions may result in a maximum stress condition, critical fatigue condition, electronic or mechanical component failure and personnel fatigue.

Under impulsive loading, the maximum response is dependent upon the relationship between the natural frequency of the structure and the time rate of change of the impulse. The rate of decay of the oscillatory response is a function of the damping. Since the natural frequency of a structure can be controlled at least to a limited degree in design and since the maximum response is a function of the relationship of natural frequency to impulse shape, it is impossible to compare structural materials or designs except in specific examples. However, it is generally agreed that reinforced plastics have a greater degree of damping than do metal structures, so that the rate of decay of the oscillatory response would be greater for the plastic structure. Therefore, assuming an equal magnitude of initial response, the plastic structure would be subjected to a fewer number of oscillations of lesser amplitude, thereby enhancing its fatigue life.

Under periodic loading, the response of a structure is primarily a function of the relationship of the natural frequency to the forcing frequency. When the two frequencies are equal, the response becomes infinite except for limitations provided by damping. As the ratio of natural frequency to forcing frequency becomes larger, the response becomes less, approaching a magnification of one, indicating a response that is equal to the forcing function. As this ratio of frequencies becomes smaller, less than one, the response becomes less, approaching a limit of zero. At frequency ratios appreciably different from one, damping has very little effect on response to this type of loading. However, at or near the resonant frequency damping is quite effective in reducing the response.

A measure of structural damping is the logarithmic decrement or rate of decay of response during free vibration. The logarithmic decrement is given by the following expression:

$$\delta = \frac{2 \pi g}{(1 + \sqrt{1 + g^2})}$$

where g is a damping factor. In metal aircraft type structures, the value of g varies from .02 to .08 Using these values in a response equation, the dynamic magnification at resonance corresponding to $g = .02$ is approximately 50 while $g = .08$ results in a magnification of approximately 12. The appropriate value of g for reinforced plastic structures is not known, but it can be seen that the magnification reduces rather rapidly with increasing values of the damping factor g.

The relatively high material damping of reinforced plastics is also quite effective in reducing noise transmission.

Under certain conditions, a disturbed elastic system may absorb energy from its surrounding media. If the energy from damping is greater than the absorbed energy, then the oscillations resulting from the disturbance will diminish with time. If the two energies are equal, then the oscillations will be maintained at a constant amplitude. Finally, if the absorbed energy is greater than the available energy from damping, then the oscillations will increase in magnitude until failure of the system occurs. These oscillations are characterized as self-induced oscillations. Flutter is an example of this phenomenon. The analysis of the flutter problem is quite complex, and it is quite difficult to predict the effect of the use of reinforced plastics on this phenomenon without considerable study.

In general, it is desirable and in many cases necessary to design structure so that its natural frequency does not coincide with primary exciting frequencies. However, in many cases it is impractical to avoid all of the exciting frequencies one hundred percent of the time. Under these conditions, the increased damping available in reinforced plastics would reduce the magnitude of the induced loads. For structures subjected to impulse or random frequency loading, the damping inherent in reinforced plastics is effective in increasing fatigue life.

The low modulus of elasticity combined with the damping makes this material effective as a shock absorber, for example, in landing gear structure.

It is concluded that the characteristics of reinforced plastics make it a desirable structural material for application in a dynamic or vibrational environment. As indicated previously, there are no quantitative data available on the magnitude of damping inherent in reinforced plastic structures. It is realized that the magnitude of damping is dependent upon the type and complexity of the structure; it is therefore desirable that further testing, to include full-scale testing, be accomplished to better evaluate this characteristic.

AERODYNAMICS

Important aerodynamic advantages can be realized through the use of glass reinforced plastics as primary aircraft structure. These advantages result from improved aerodynamic cleanliness or shape due to the following characteristics of reinforced plastics:

1. Improved surface finish inherent in plastics.
2. Elimination of surface imperfections such as rivets, gaps and lap joints by use of integral and/or bonded structure.
3. Smoother contours free from local deformations and wrinkles by use of stable monocoque construction.
4. Improved aerodynamic shape due to the greater rigidity inherent in some types of reinforced plastic structures.

The improvements contributed by these items result in reduced aerodynamic drag and increased lift characteristics, thereby providing potential increases in speed, range and economy of operation.

Skin drag is the product of the surface shear, developed by moving a body through a viscous medium, and the surface area. The shear value is greatly influenced by the nature of the boundary layer surrounding the moving object. This boundary layer will either be laminar, characterized by a small velocity gradient and producing low shear, or turbulent, characterized by a thickened boundary layer and a large velocity gradient producing high shear. Deterioration of laminar flow characteristics and transition to turbulent flow may result from such effects as operation in turbulent or hot air, from vibration or noise, or from disturbed flow brought about by surface irregularities. It is the latter of these disturbing elements that can be appreciably altered through employing construction techniques embodying plastics and bonded structures. Some of the surface irregularities common to sheet metal construction but eliminated through plastic construction are rivets, lap joints, gaps, and "normal" fabrication skin roughness or irregularity.

When an airfoil, for example, is sufficiently rough to cause transition from laminar to turbulent flow near the leading edge of the section, large increases in drag are incurred. This effect is clearly seen in Figure 2, which shows the variation of drag with surface condition and Reynolds number. In subsonic flow, well below the acoustic velocity, the variation in fluid density may be neglected, so the flow conditions and drag are functions of Reynolds number, R, where

$$R = F(\rho, V, l, \mu) \text{ or } R = \frac{\rho V l}{\mu}$$

where ρ = fluid density constant
V = velocity of object
l = reference length
μ = coefficient of absolute viscosity

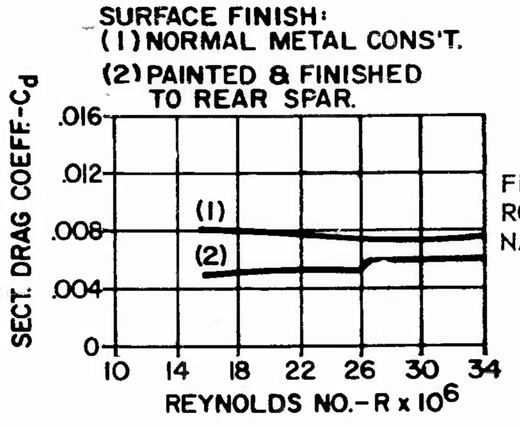


FIGURE 2. EFFECT OF REYNOLDS NUMBER AND SURFACE ROUGHNESS ON DRAG COEFFICIENT FOR AIRFOIL SECTION NACA 65(216) - 3(16.5) C

Tests have been conducted in the NASA Langley two-dimensional low-turbulency pressure tunnel in order to compare typical practically-constructed metal airfoil sections with those of varying degrees of smoothness. Results of these tests conclude that smooth surfaces always produce substantial drag reductions. Figure 3 shows a comparison of a smooth and roughened leading-edge airfoil section. The roughness (0.011-inch grains) is more than the usual manufacturing irregularities, although less than for accumulated ice and mud which are occasionally encountered in regular operation. It does, however, indicate the seriousness of surface roughness and points up the desirability of close control of surface conditions. In fact, surface quality was found to have more effect on the minimum drag characteristics than the type of airfoil section.

Generally, in subsonic flow, sections of "practical" construction produced a drag coefficient between 0.007 and 0.008 in nearly all cases, regardless of section. The data also showed that airfoils permitting extensive laminar flow had substantially lower draw coefficients when smooth than those with limited laminar flow. Once sufficient roughness was present to force transition from laminar to turbulent flow, additional roughness produced very little added effect. The degree of roughness was shown to have a much larger effect on drag at high lift coefficients. A supplementary effect of leading edge roughness is to decrease the lift curve slope, particularly for thick sections having the position of minimum pressure far back on the section.

Reinforced plastic construction, with its inherently smooth surfaces, will provide flight articles requiring less thrust with accompanying increase in range at the same airspeed, or will provide an increase in cruising speed with the same range.

Research and development are currently underway in the field of boundary layer control designed to delay or prevent flow transition over the entire aircraft surface for maximum aerodynamic benefits. Predictions of subsonic performance gains utilizing 100 percent boundary layer control indicate extremely large benefits to be obtainable. Achieving these goals will require smooth, close tolerance contours that can be best provided by glass reinforced plastic and bonded type construction.

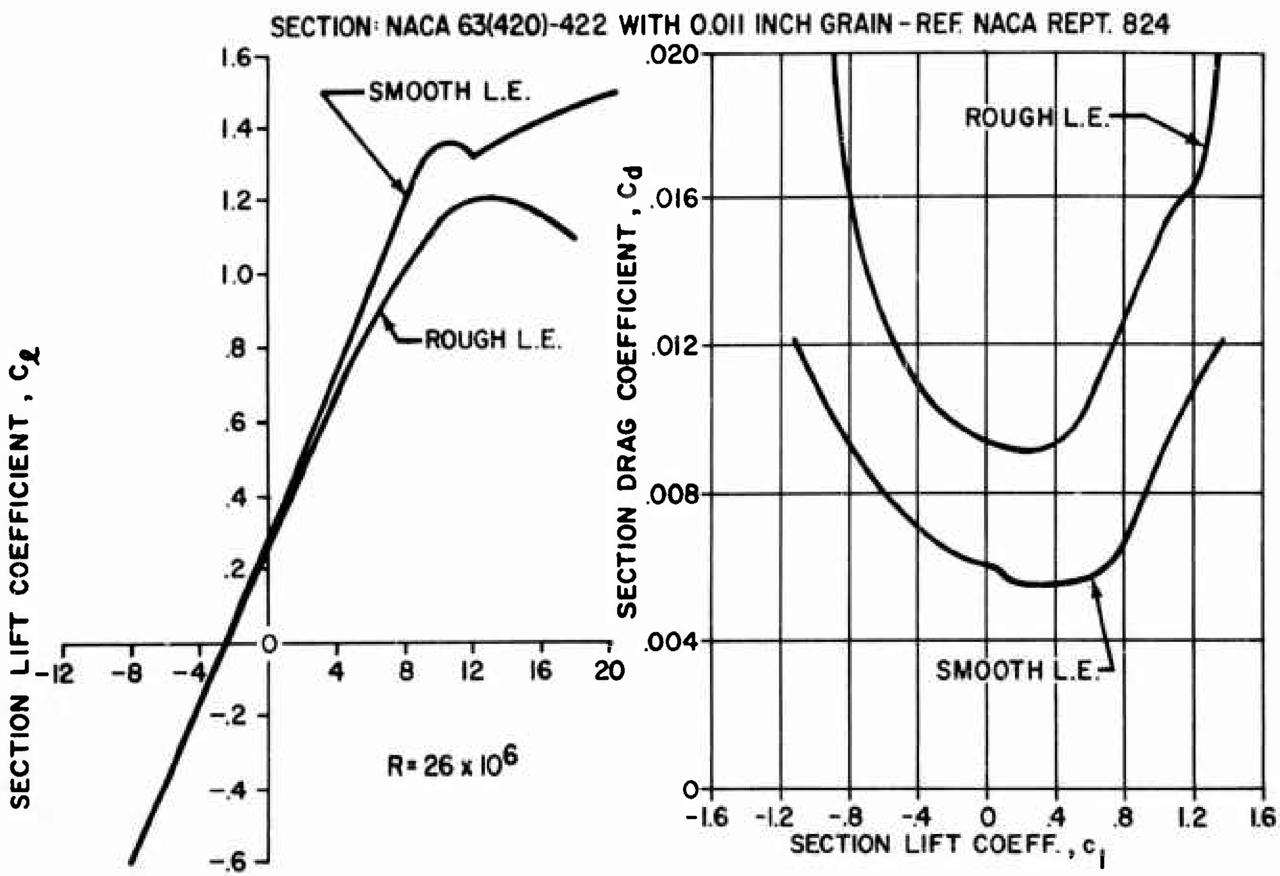


FIGURE 3. TYPICAL EFFECT OF ROUGHNESS ON LIFT AND DRAG CHARACTERISTICS

A number of research and development programs have been conducted utilizing boundary layer control for improved vehicle lift/drag relationships and reduced power requirements. Foremost in the application of boundary layer control to light aircraft has been the Mississippi State University. Test beds for past research and development include the Beech D-18 and AT-11 multiengined aircraft and the Army L-19 and L-23 light aircraft. Modification of these aircraft generally encompassed several techniques for improving performance in addition to boundary layer control application. Each technique relies heavily upon smooth surfaces for aerodynamic efficiency. For each vehicle modification, all surfaces were smoothed where possible, protruding rivets covered, fillets introduced in such areas as the wing root and nacelle juncture, low-drag wing tips installed, canopies smoothed and external protuberances suppressed and faired. Such modification, for example, enabled the L-23 Twin-Bonanza to cruise at 190 miles per hour on 58 percent full throttle horsepower in contrast to its original maximum speed of 187 miles per hour with full throttle.

Mississippi State is currently engaged in the development of a boundary layer control two-place, 90-horsepower, light plane called the "Marvelette", with anticipated performance enabling take-off and landing at 35 miles per hour and a top speed of 200 miles per hour. This aircraft incorporates reinforced plastic wings, fuselage, nose section and ducted fan shroud.

Concurrent with the "Marvelette" development, Mississippi State is developing a boundary layer control two-place Army vehicle which has an Allison T63 turboprop engine that develops 250 horsepower. This aircraft, called the "Marvel", provides for engine intake air to be sucked through perforations in the wing upper-surface skin. To make this system efficient, the skin will be molded fiberglass so that all surfaces will have maximum smoothness, thereby eliminating unnecessary drag and providing an excellent surface for boundary layer control application. The "Marvel" is all reinforced plastic construction.

It is concluded that the extremely smooth and accurate contours, routinely attainable using glass reinforced plastics construction, will result in significantly improved aerodynamic characteristics of aircraft employing such materials over appreciable areas of structure exposed to the air stream. Wing lift values as much as 20 percent higher than those of average conventional metal construction and profile drag values as much as 20 percent less are readily obtainable by exploiting the improved aerodynamic surfaces of reinforced plastics. Equivalent advantages may accrue as the use of smooth finish plastic materials and practices are extended to other areas of aircraft. It is possible to multiply these improvements several times in cases where the smooth surfaces are teamed with effective boundary layer control techniques.

In certain applications, reinforced plastics have another inherent capability for improving aerodynamic efficiency. This exists where plastic construction lends itself more readily to streamlined shapes, due to ease of fabrication, than does conventional metal construction. Typical examples are landing gear struts and miscellaneous protuberances which can have minimum parasite drag through use of clean, faired and smooth configurations.

Improved range, speed, and payload or combinations of these parameters will result from drag reduction or lift improvement attendant upon the substitution of reinforced plastics for conventional metal construction. The extent or degree of enhancement of desirable aerodynamic qualities will require evaluation in each particular instance, but it is certain that, in many cases, such evaluation will justify considerable added cost.

DESIGN STUDIES

Design studies of typical current and projected Army aircraft components in the following categories have been investigated:

1. Fuselage
2. Landing Gears
3. Transmission Housings
4. Drive Shafting
5. Empennage
6. Fuel Tanks
7. Wing
8. Rotor and Propeller Blades

The feasibility of embedded electrical conductors and hydraulic pipe in laminates, problems of rain erosion, and the compatibility of reinforced plastics and hydrocarbon fuels were investigated.

The approach to each design study was based on preliminary evaluation of the nature of the particular problem areas and the state of development of known reinforced plastic components. For example, in the case of rotor blades, much highly specialized development work has been accomplished, the magnitude of which is much greater than this entire program; the effort in this area was therefore directed at determination of and evaluation of the work accomplished by others.

In the case of empennage applications, typical configurations were studied in detail and several alternate reinforced plastic designs have been evaluated.

All of the design studies have been based on current and projected Army requirements. The reinforced plastic designs generated are consistent with applicable criteria and conventional aircraft practice. Where appropriate, existing conventional Army aircraft structures and components were used as a basis for the reinforced plastic design studies so as to provide a comparable conventional metal design. This procedure precluded the needless expenditure of time in generating conventional designs for comparison. The applicability of the conventional design to the study requirements was of course verified. However, it should be noted that it was not possible to optimize the reinforced plastic designs during this study to the same degree that existing production metal counterparts have been optimized. Therefore, the reinforced plastic designs are at some disadvantage in comparison.

In the following sections of this report, the various design studies are presented in summary form. Sufficient pertinent detail is presented to define the coverage of each study and to substantiate the conclusions.

In general, each design study was conducted in the following manner:

1. The requirements for the particular structure or component were studied.

2. Existing Army aircraft applications, projected future applications and related applications were studied for selection of the primary areas of interest.

3. Where appropriate, existing designs were selected as a basis for the reinforced plastic design studies and subsequent comparative evaluation.

4. Potential reinforced plastic design and fabrication approaches were developed and subjected to preliminary evaluation, and the most promising was selected for detailed design study.

5. Each design study configuration was optimized to the degree possible in this limited program; evaluated with respect to each pertinent engineering, fabrication and service parameter, including cost; and compared to the existing conventional design. Advantages and disadvantages, conclusions and recommendations were summarized.

Cost evaluation of the various reinforced plastic designs is a very vital part of the feasibility study. Arriving at reliable cost figures, however, appears to be a most difficult task. It is apparent that many of the hard facts necessary for accurate cost estimating are quite elusive or nebulous. This is primarily due to the lack of industry experience in reinforced plastic structures and components of the types considered in this investigation. It follows, of course, that actual cost data are nonexistent.

Cost estimating capability is built primarily on experience. Where direct experience is not available, it is desirable to average out the potential error by accruing estimates from several sources engaged in related work. This procedure has not produced results, however, due to a lack of interest by most of the fabricators contacted.

Regardless of these handicaps, other difficulties related to product optimization must be considered. The great variety of methods and techniques in plastics design and fabrication presents innumerable approaches to minimum cost. Here again, lack of industry experience precludes the short cuts to cost estimating reinforced plastics.

In view of these problem areas, the current study evaluation of cost was based primarily on estimates by Hayes Cost Analysts. This approach insured consistent information and probably provides the best basis for comparative evaluation of different reinforced plastic alternatives.

The remaining problem involves comparison of reinforced plastic and conventional metal configurations. Available cost information for existing components is essentially the "spares" cost of a developed product as taken from the Federal Stock Catalog. Such data should reflect the ultimate in low cost. In order to present a fair comparison, these costs should be verified since there appear to be inconsistencies in some cases. Such verification has not been possible to date.

Under any circumstances, the costs quoted for reinforced plastic components should be regarded as approximate and tentative. Any apparent disadvantage of reinforced plastics versus conventional metals must be tempered with the realization that the conventional configuration has had the benefit of a much higher degree of optimization.

In order to avoid repetition in the presentation of the design studies, discussions of the various technical problem areas have preceded this section of the report. For the same reason, the general advantages and disadvantages of reinforced plastics compared to the conventional metals are summarized below. Therefore, it will not be necessary to repeat these points in the various design study evaluations which follow.

Advantages

1. Broad choice of material properties, characteristics, and fabrication processes for configuration optimization.
2. High strength/weight ratio.
3. Improved mechanical and acoustical vibration damping.
4. Improved energy absorbing capability in the elastic range.
5. Simplified integral design and fabrication resulting generally in lower cost.
6. Reduced maintenance; noncorrosive, durable, easily repaired.
7. Improved aerodynamic efficiency through surface smoothness and contour.
8. Nonmetallic/noncritical material.
9. Radar transparent; electrical and thermal insulators.
10. Less vulnerable to small-arms fire.

Disadvantages

1. Less design and fabrication knowledge available, necessitating development for most applications.
2. Quality more sensitive to process control.
3. Potential development costs to optimize processes and product applications.
4. Higher cost raw materials.

FUSELAGE DESIGN STUDY

This design study covers the utilization of reinforced plastics as a basic material for the fabrication of aircraft fuselages. Potential Army aircraft fuselage applications cover a great range of types, sizes, and shapes. Existing and anticipated future configurations have been reviewed in order to concentrate the study effort in those areas likely to best satisfy the overall intent of the program.

Fuselage design differs from the design of other primary structure of an aircraft, particularly in the relative complexity of the requirements. A rotor blade may be said to be essentially 100 percent primary structure adapted to carrying air loads and those loads imposed by centrifugal force. Similarly, a wing, landing gear, tail surface, and the tail boom portion of a fuselage devote a large majority of their structure to resisting "primary" loads; that is, flight air and inertia loads or ground landing loads. A great preponderance of the structural design effort is directed toward provision of structure to resist these primary loads.

The average fuselage, on the other hand, contains fully as much "secondary" structure as primary. In many instances, accommodations of the secondary items have a considerable influence on the primary structural areas. The secondary structural items referred to are such things as doors; windows; seats for passengers and flight crew; litters; provisions for support and tie-down of cargo; and support for and enclosure of electronic, first aid, oxygen, flotation gear and parachute equipment. In addition, if armament is fitted, adequate structure must be provided not only to support the equipment but also to resist recoil, antirecoil and muzzle blast loadings.

Successful solution of the detail structural and mechanical problems associated with incorporation or installation of these and allied items of equipment is as much a part of a successful fuselage design as are the problems of reacting primary flight or landing loads. Unless or until such secondary items which form so much of the typical fuselage are provided for, it is not practical to attempt a true comparison of the reinforced plastic design with one of conventional materials. This is not to imply that the existence of such problems should in any way inhibit the application of glass reinforced plastics in this area. It is felt, however, that such a comprehensive design effort is not within the scope of the present investigation; and therefore the study of complete fuselages would not be truly productive and is not warranted at this time.

In view of the magnitude of the overall fuselage design problem, it has been deemed most appropriate to aim the current study at the helicopter tail boom problem area. This type of structure is visualized for many future applications, including the light observation helicopter (LOH) now under development.

In current helicopter configurations, the tail boom structure falls into two general categories. One configuration consists of semimonocoque structure while the other is an open truss type structure, both of which

utilize aluminum and magnesium alloys. In many cases, the working stresses are quite low to avoid problems of local instability or merely to maintain gages and sizes that have sufficient durability against secondary loads. Because of the very low damping available in metal structures, care must be exercised to avoid certain critical frequencies, and in many cases the design is dictated by this condition.

A preliminary strength-weight comparison of the usual sheet-metal type of construction and the possible reinforced plastic types of construction indicates that static strength requirements can be met with the plastic design with equal or less structural weight when compared to the metal design.

The effect of comparative stiffnesses of the metal vs. plastic design is not as easily analyzed. In the general case, equivalent geometrical designs in aluminum and plastic to equal strength will result in a plastic structure that is more flexible than the aluminum one. However, it may be just as simple to avoid critical frequencies with one as the other. Furthermore, with the increased damping inherent in plastic design, it may be permissible to operate at or at least nearer to critical exciting frequencies without incurring undue magnification of vibratory loads. In addition, the greater freedom of geometrical design afforded by reinforced plastic techniques will produce more efficient use of the mechanical properties of the material.

In consideration of the helicopter tail boom problem areas, it appeared desirable to study typical configurations in both the medium and light helicopter categories. This is due to the potential variation in degree of feasibility with size and complexity of design. For study, the HU-1 and the H-23 models were selected as representative of typical applications in these categories. In addition, these two aircraft employ configurations similar to projected LOH models and other potential future applications.

Solid laminate and sandwich construction were investigated for both tail booms. Type 181 fiberglass cloth impregnated with epoxy resin was used in the analysis for the solid laminates and for the sandwich faces. There are several reasons for choosing the 181 glass cloth and epoxy resin. Panel instability is the critical failure mode of this particular structure. The failure stress in this mode is a function of the product of the moduli of elasticity in the direction of the load and perpendicular to the load. The 181 cloth provides a relatively high value of this product. Since the direct stresses are relatively low, there is no requirement for a high concentration of fibers in any particular direction. Reliable data are available for this type of cloth, and since no other cloth or fiber presents a major advantage for this application, 181 cloth provides a logical basis for preliminary analysis.

Other materials deserve consideration for these applications. The new high-strength unidirectional nonwoven fabrics such as Minnesota Mining and Manufacturing Company's "Scotchply" will have definite advantages in specific applications. It can be obtained in unidirectional, crossplied,

and isotropic fabric. The mechanical properties are somewhat greater than those for woven fabric. The test program for this study included some evaluation of "Scotchply".

Hercules Powder Company has accomplished some experimental work in molding parts with their "Spiralloy" mat. This mat is a filament winding of any thickness wound on a large-diameter mandrel, then split and removed from the mandrel in the "B" stage of cure. It can then be handled and molded the same as other preimpregnated cloth. The primary advantage of "Spiralloy" mat is its high strength and exceptionally good drapability. Some data on this material are included in the test section of this report. It appears to have excellent potential for faces of sandwich construction of intricate shapes that require high strength.

The analysis for sandwich construction was based on the use of fiberglass honeycomb core. The primary advantage of the fiberglass core is its radar transparency. It would probably provide better damping characteristics than the aluminum honeycomb, but much additional testing is required to determine the magnitude of the damping involved and to evaluate the overall effect on the design. Other core materials are easier to use and less expensive in fabrication. Aluminum honeycomb, paper honeycomb, foams, Narmco "Multiwave", and "Trussgrid", by General Grid Corporation, were considered. A complete evaluation of all these materials could not be accomplished in this program. Any of them can be used to obtain a satisfactory structure, and the analysis would be essentially the same.

Aluminum honeycomb is considered to be representative of the complexity of fabrication, weight and cost. It is considered to be the optimum at the present time, and the evaluation of the components is based on its use.

Aluminum honeycomb can be machined in the unexpanded condition and has moderate forming characteristics. This is an advantage over fiberglass honeycomb, which must be machined in the expanded condition and is difficult to form. "Multiwave" and "Trussgrid" offer better forming characteristics but will be slightly heavier.

Plastic foam offers a more nearly continuous support for the faces; however, this advantage is counteracted by the fact that the foam has a much lower modulus of elasticity, so that the net result is in doubt. Although the damping qualities of the foam core are probably better than either the aluminum or fiberglass honeycomb, the resistance of the foam sandwich to vibration is somewhat questionable. In general, a great deal of additional test data are required to permit a good evaluation of the possible use of foam sandwich for this application.

The test program included compression and bending tests of sandwich panels with aluminum honeycomb, "Multiwave", and polyurethane foam. "Trussgrid" is a relatively new material and information was received too late to be included.

HU-1 TAIL BOOM

The HU-1 tail boom is a conventional skin-stringer-frame semimonocoque design of aluminum and magnesium alloys. It supports a horizontal control surface and also a fixed vertical stabilizer to which is mounted a tail rotor.

Design data have been taken either from the helicopter itself or from the limited number of drawings available on the tail boom. The weight distribution and design criteria are based on data in Bell Helicopter Corporation Report No. 204-947-035, "Detail Specification for HU-1A Utility Helicopter" (Reference 13). The loading condition assumed in the analysis combines a 1220-pound tail rotor thrust with 1.8 g. limit gust load factor. A 380-pound down load is assumed to be acting on the horizontal stabilizer. The tail rotor thrust force is derived from Bell Helicopter Corporation Report No. 204-099-753, XH-40 Stability and Control Analysis" (Reference 15).

The geometry for the reinforced plastic boom is arbitrarily chosen to be essentially the same as the existing metal boom. For this geometry and for the magnitude of the loads involved, a sandwich type construction proves to be most feasible. A skin-stringer-frame type of construction requires close spacing of stringers and frames to maintain stability of the structure; it therefore involves assembly of many pieces, thereby losing the economic advantage of large-scale molding techniques of fabrication. For a pure monocoque of simple laminated construction, the thickness required for a panel stability for the cross sectional dimensions involved makes the weight prohibitive.

The methods of determiing sandwich panel buckling allowables used in the stress analysis for the tail boom study are presented in Forest Product Laboratory Report No. 1867,"Compressive Buckling Curves for Simply Supported Sandwich Panels with Glass-Fabric-Laminate Facings and Honeycomb Cores," Reference 56. Theoretical panel buckling data are shown in Figure 4. The curves show the predicted buckling stress for 22-inch and 31-inch wide panels versus panel thickness for various face thicknesses. The horizontal lines represent the face buckling stresses for the various face thicknesses. The panel sizes were used as being equivalent to the large radius curved panels of the tail boom - the larger size for forward end and the smaller size for aft sections. Since no test data were available to substantiate the theoretical data, flat and curved sandwich panels of various radii fabricated of several materials were tested. The results of these tests substantiate the choice of materials and sizes for the HU-1 tail boom. A further discussion of the tests and detail results is included in the Test Section.

The shear and bending moment curves for the loading conditions are shown on Figure 5. Automatic computation was utilized to calculate the section properties and bending stresses for various face thicknesses. By comparing these calculated stresses to the allowable panel buckling and face crippling stresses, the appropriate combination of panel depth and face thicknesses was determined.

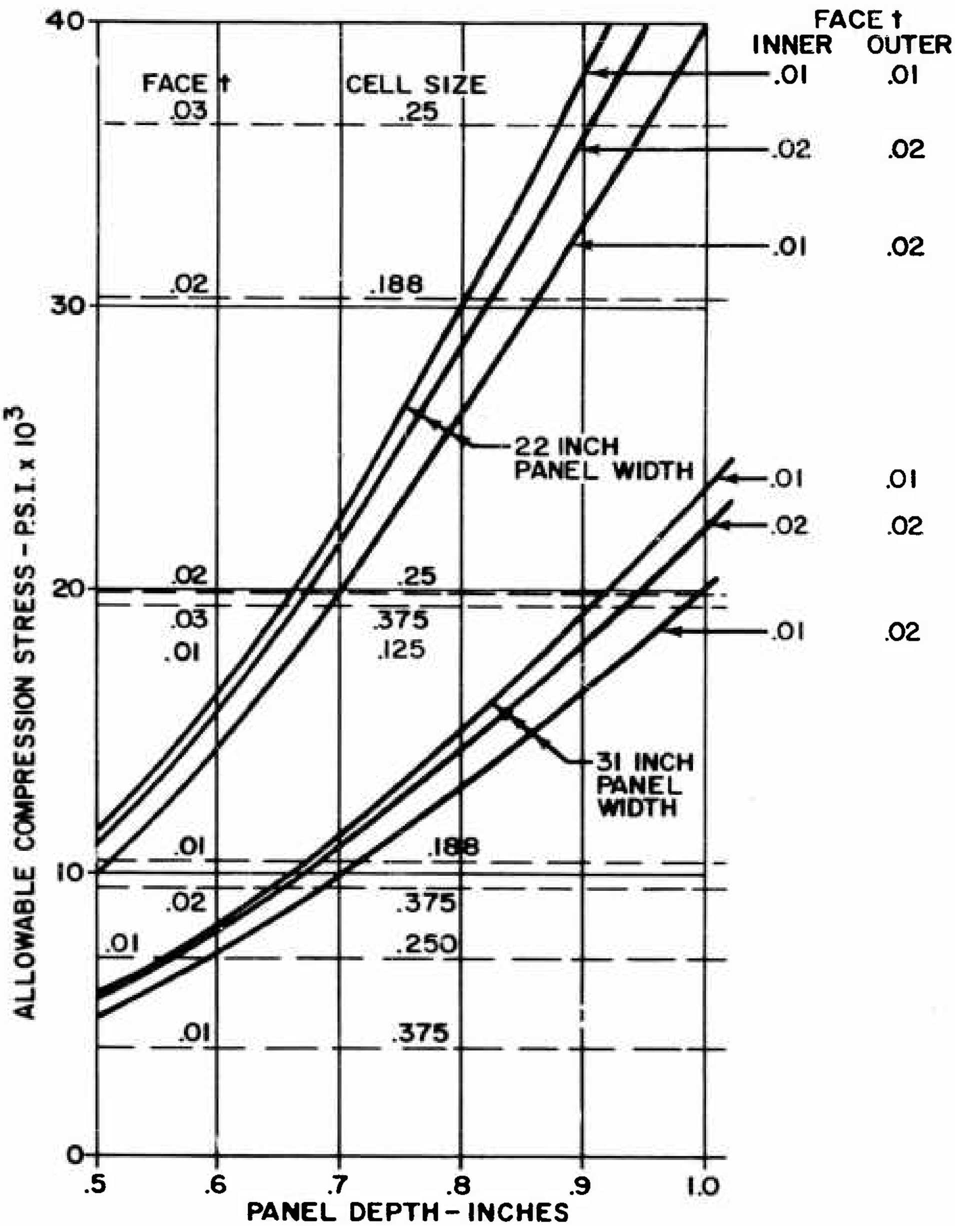


FIGURE 4. THEORETICAL BUCKLING STRESSES FOR REINFORCED PLASTIC SANDWICH

The required panel depth is based on an analysis of the sides of the boom, which are relatively large panels with a very large radius of curvature. It is recognized that around the top and bottom of the boom an appreciably lesser panel depth could be used. Without performing a detailed analysis, it is estimated that this depth could be approximately 1/4 inch. This form of taper requires additional machining of the core material but simplifies the forming process and reduces the weight.

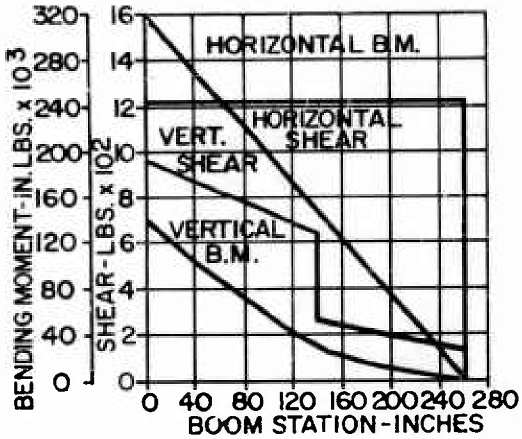


FIGURE 5. HU-1 TAIL BOOM, ULTIMATE SHEAR AND BENDING MOMENT

Three reinforced plastic configurations are suggested as representative of feasible ways in which the HU-1 tail boom can be fabricated. Each is a variation of essentially the same structure but each has some features that may prove to be advantageous in achieving the optimum.

Configuration I, shown on Figure 6, is a one-piece structure that is fabricated in its entirety on a male mandrel. Configurations II and III, shown on Figures 7 and 8, utilize the same type of structure but are fabricated in two sections. Configuration II is split along the vertical centerline and Configuration III is split along the horizontal plane of maximum width. The fin can be fabricated along with the body section in Configuration II, as shown in Figure 9. Although this is a potentially feasible method of fabrication, it is believed that some problems will be encountered in fitting a required fin spar between the two body-fin sections due to dimensional variations. This composite structure should be evaluated for any similar new design. Integrally molding components together aids in realizing the full potential benefits of reinforced plastic structures. When reinforced plastics are given serious consideration for a new design under development, the configuration can perhaps be modified to accomplish maximum composite fabrication with economy.

The metal boom of the HU-1 attaches to the forward body at four points. To maintain this same four-point attachment in the reinforced plastic boom, four fittings and local area reinforcements must be provided to distribute the splice loads into the sandwich monocoque structure. In a new application, it would be particularly advantageous to utilize a continuous-type connection at the splice.

An analysis of the requirements for detail attachments, for example, attachments of the drive shaft bearing blocks to the boom, is not presented. However, some means of distributing the loads at these points into the boom structure must be supplied. Possible ways of doing this are indicated in the sketches. It could also be done by utilizing pre-

formed sections, for example, channels or hat sections within the sandwich panel.

The existing metal tail boom of the HU-1 weighs 124.5 pounds (Reference 13, Bell Report No. 204-947-035). The basic skin and honeycomb structure of the plastic design weigh approximately 80 pounds. The difference of approximately 45 pounds is considered more than adequate to provide for such things as end closures, local reinforcements, drive shaft cover, etc., making the reinforced plastic competitive with the metal weight-wise in meeting strength requirements.

The bending natural frequency of the plastic boom is computed as 334 cycles per minute in a vertical plane and 281 cycles per minute in a horizontal plane. The operating range for the main rotor is 280 to 315 r.p.m. Since there are no quantitative data available on damping of reinforced plastic materials, it is not known whether or not this is a safe operating condition. With some sacrifice in weight, the natural frequency could be increased as required. However, it is probably more desirable to modify the geometry of the tail boom. By reducing the cross-sectional dimensions, panel buckling allowable stresses would be increased perhaps enough to accommodate the increased stresses resulting from the section change. In this manner, it may be possible to reduce the natural frequency sufficiently below the exciting frequency to provide an even more favorable condition.

Cost analysis of the reinforced plastic configuration is given in Table I.

TABLE I
COST ANALYSIS OF HU-1A TAIL BOOM

	Unit Cost	
	Quantity of 10	Quantity of 100
Configuration I	$ 6820	$ 2640
Configuration II	4915	1995
Configuration III	5940	2125

Federal Stock Catalog "Spares" cost of the existing hardware is:
HU-1A Tail Boom $ 2890

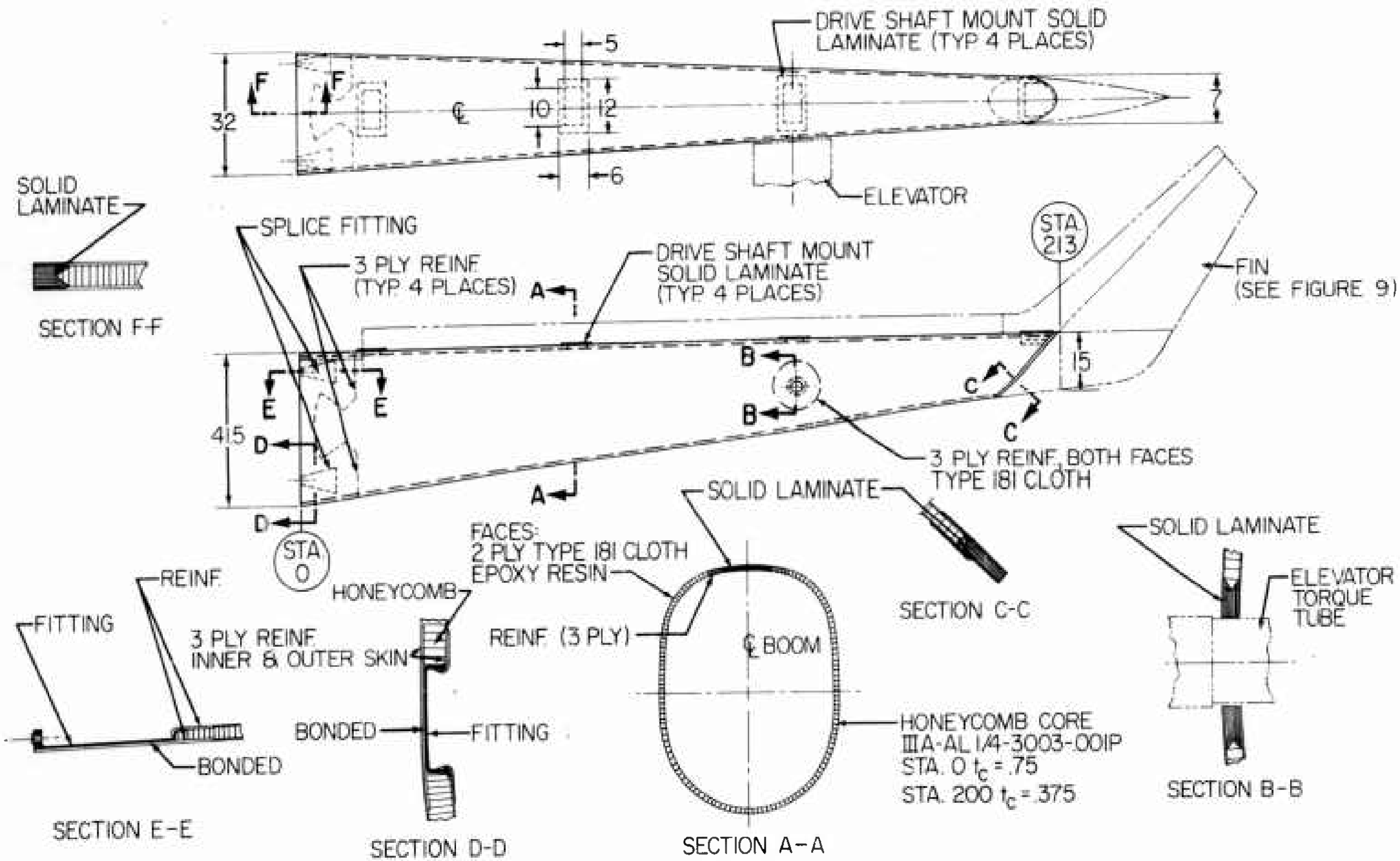


FIGURE 6. HU-1A AFT BODY OF REINFORCED PLASTIC, CONFIGURATION I, ONE PIECE SANDWICH

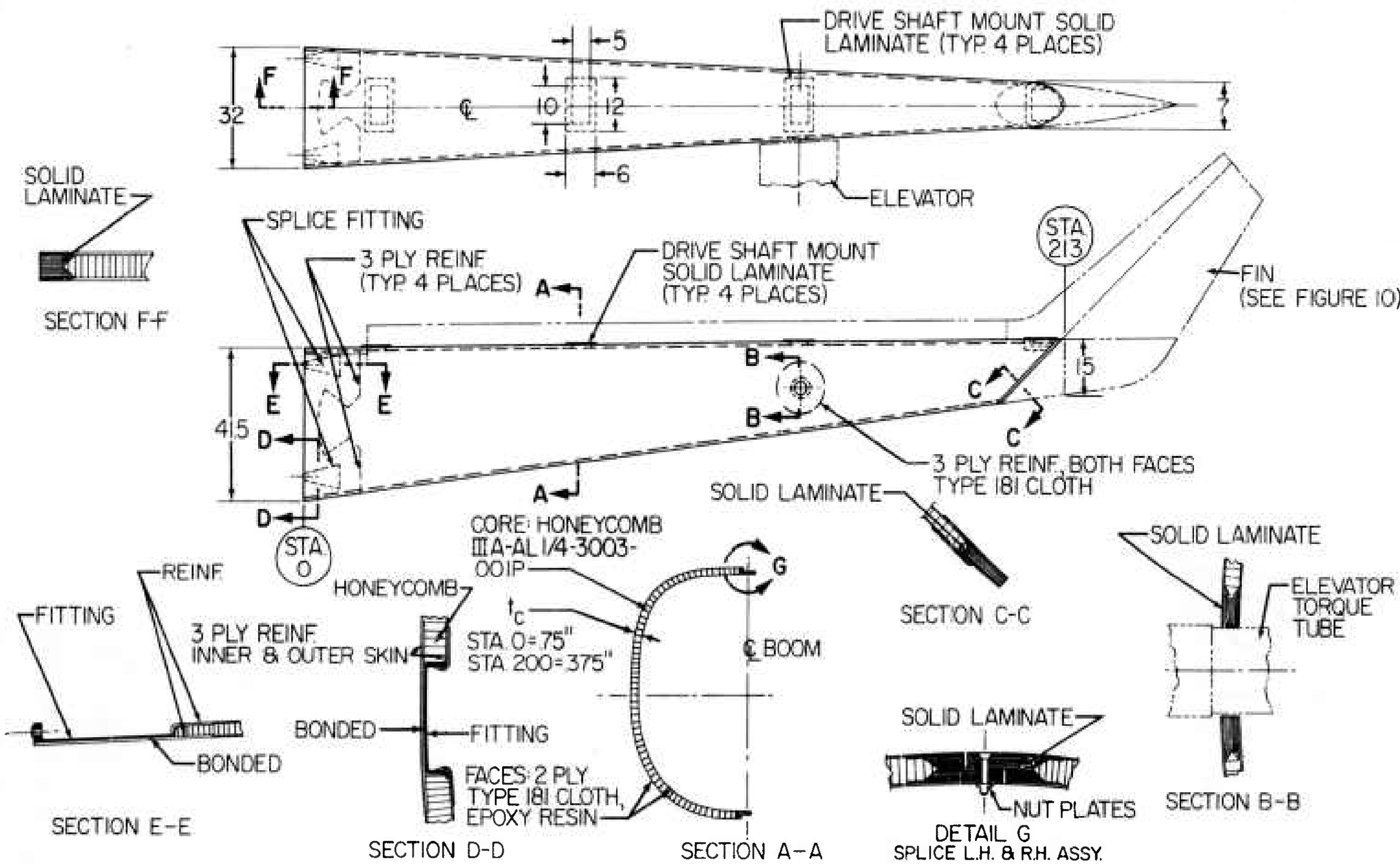


FIGURE 7. HU-1A AFT BODY OF REINFORCED PLASTIC, CONFIGURATION II TWO PIECE SANDWICH, SPLIT AT VERTICAL CENTERLINE

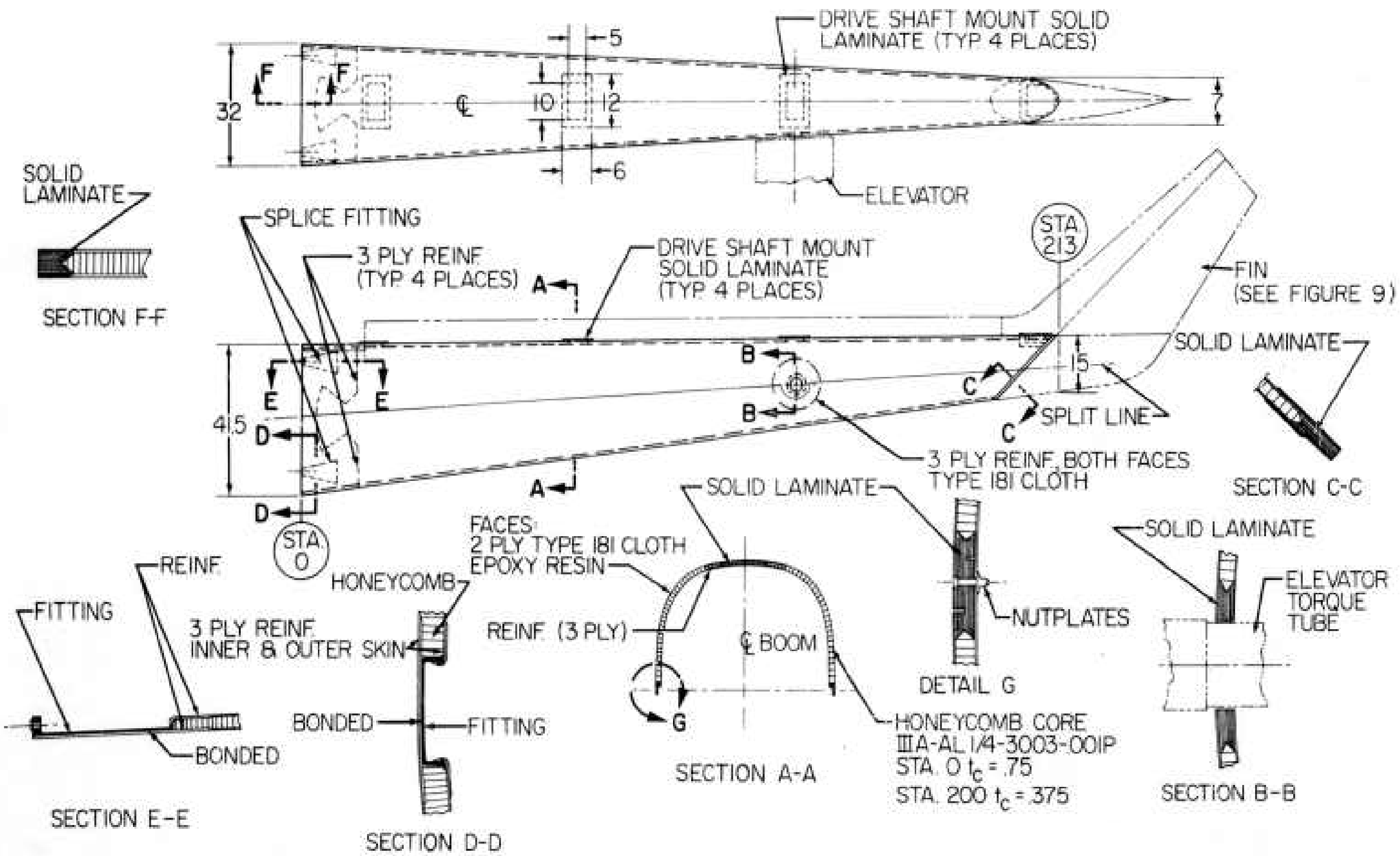


FIGURE 8. HU-1A AFT BODY OF REINFORCED PLASTIC, CONFIGURATION III
TWO PIECE SANDWICH, SPLIT AT HORIZONTAL CENTERLINE

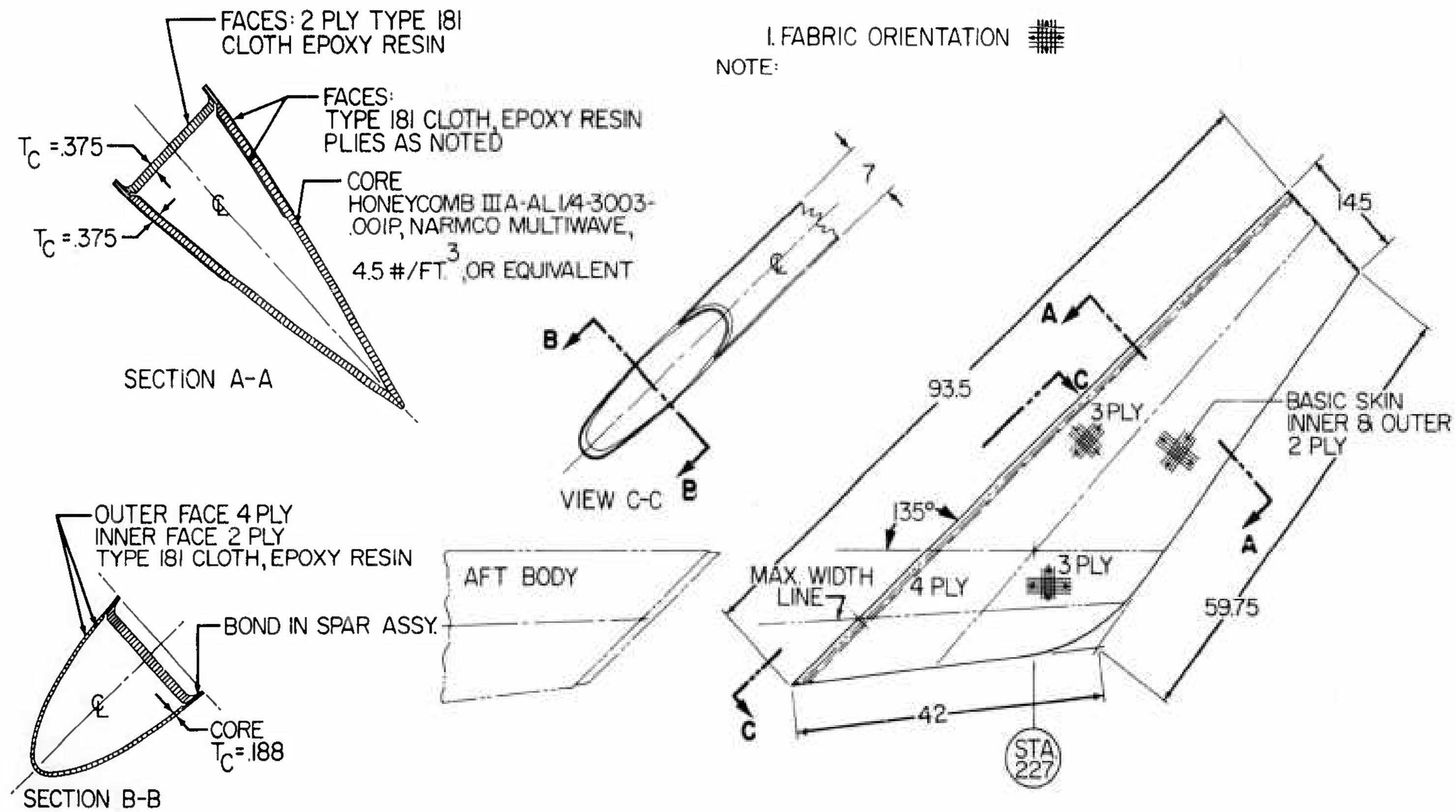
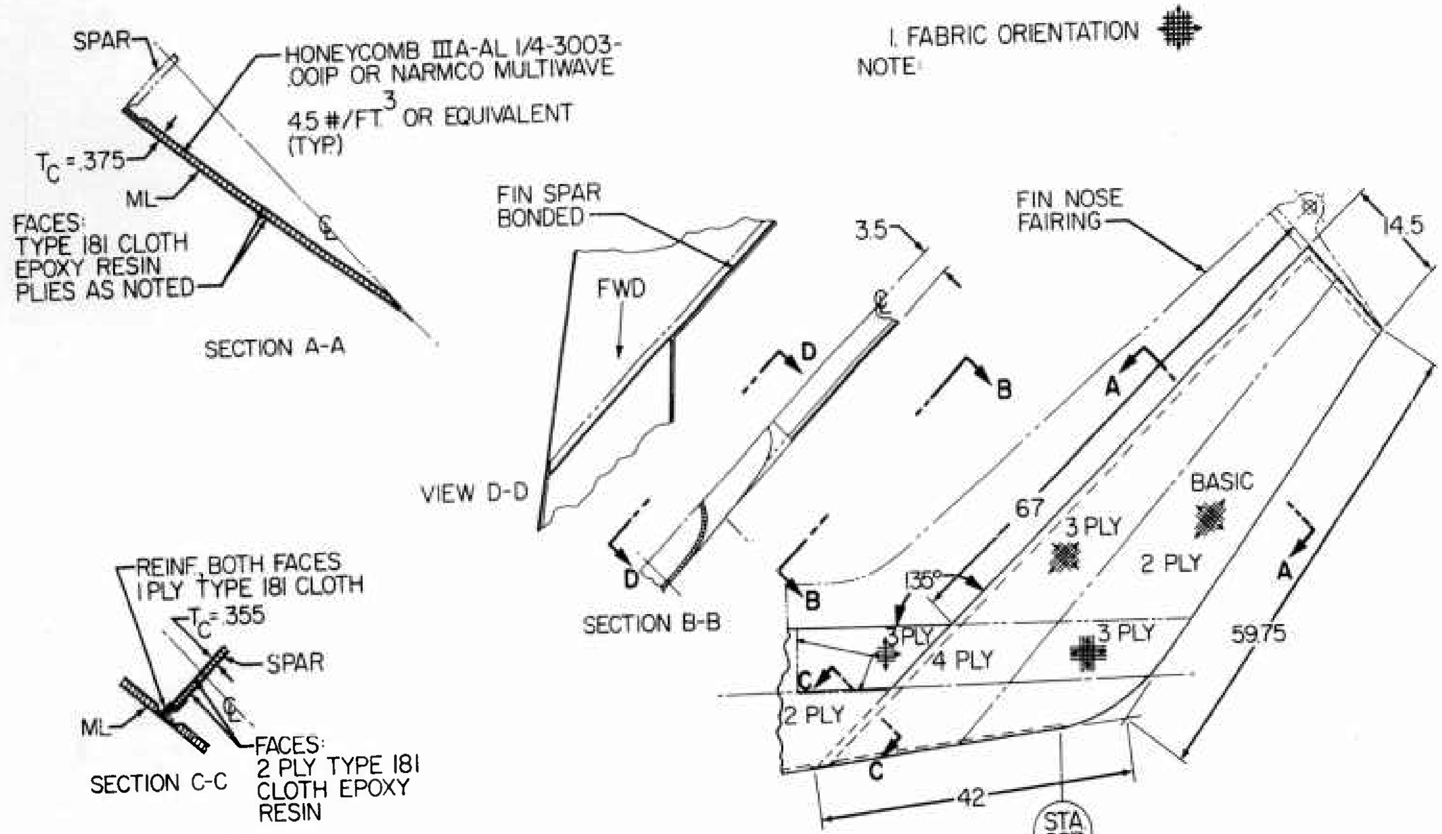


FIGURE 9. HU-1A VERTICAL TAIL SURFACE OF REINFORCED PLASTIC SANDWICH, USED WITH AFT BODY CONFIGURATIONS I & III



SECTION C-C

FIGURE 10. HU-1A VERTICAL TAIL SURFACE OF REINFORCED PLASTIC SANDWICH USED WITH AFT BODY CONFIGURATION II

H-23 TAIL BOOM

The existing metal tail boom of the H-23 helicopter is a semimonocoque aluminum structure. The overall geometry is unchanged for the reinforced plastic design study. Two types of structure are analyzed and evaluated; the first type is a pure monocoque of sandwich construction while the second is a pure monocoque of plain laminated construction. Since the boom is of a circular cross section of relatively small diameter, the plain laminated structure warrants consideration.

The loading condition used in the analysis consisted of a 5.25 g. vertical load factor applied to an estimated weight distribution combined with a 100-pound down load on the horizontal stabilizer and a 400-pound tail rotor thrust load. The tail rotor thrust load is verified by Bell Helicopter Corporation Report No. 47-030-018 (Reference 12), covering a machine of similar size, "Basic Design Criteria Model 47E".

The method of analysis presented in Forest Products Laboratory Report No. 1867 (Reference 56) is used for determining panel buckling allowables for sandwich panels. Face crippling allowables were computed for the HU-1 and are shown in Figure 4. The shear and bending curves for the loading condition are shown on Figure 11.

The allowable buckling stresses for the solid laminate construction are derived by formulation as outlined in MIL-HDBK-17 (Reference 38). The shear and bending loads for the solid laminate are basically the same as for the sandwich construction. The effects of the small difference in dead weight are negligible. Therefore, the curves as shown by Figure 11 are applicable.

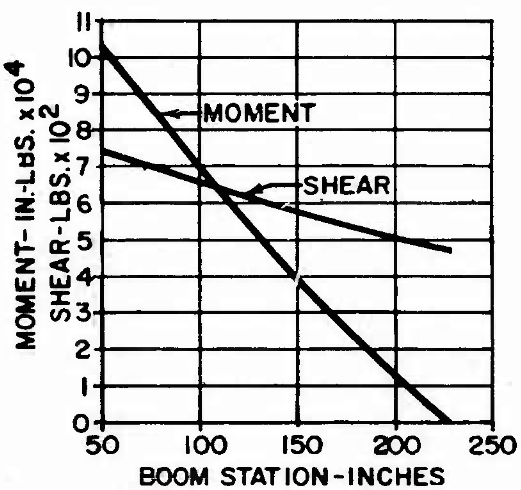


FIGURE 11. H-23 TAIL BOOM ULTIMATE SHEAR & BENDING MOMENT

Three configurations for sandwich construction of the H-23 tail boom are presented. Configuration I is a one-piece sandwich as shown in Figure 12. Configuration II, in Figure 13, is a similar sandwich but split along the horizontal centerline. An entirely different concept of sandwich construction is presented as Configuration III (Figure 14). This utilizes an integrally woven three-dimensional fluted core referred to as "Raypan", a trade name for the material recently developed by Raymond Development Industries, Inc. Additional face plies of cloth can be added to the core as required. According to the manufacturer, it can be molded to conform to contour. It appears to be

a feasible material for components of this type, but insufficient information is available to accomplish adequate evaluation relative to other materials. Sketches of a possible design using "Raypan" and assumed loads for the H-23 tail boom were forwarded to Raymond Development Industries, Inc., for their comments on this application. They considered the application to be feasible and to cost relatively little to fabricate.

A one-piece solid laminate monocoque shell, Configuration IV, Figure 15, is the most economical method of construction for a tail boom of this type. A variation of the solid laminate shell is Configuration V, Figure 16. It is similar to Configuration IV except that the boom is split on the horizontal centerline, formed in two parts, and spliced by bonding and mechanical attachments. This permits forming in an open mold with the mold surface being the external surface. By bonding the reinforcements for the torque tube mounts and tail skid in place after molding the shell, the two halves become identical and can be formed in the same mold.

The weight of the sandwich type boom is approximately 20 pounds, while the weight of the solid laminate boom is approximately 32 pounds. This is compared to an estimated weight of the existing H-23 metal boom of 30 pounds. The maximum deflection of the sandwich type boom is 11.3 inches compared to 6.4 inches for the solid laminate. The natural bending frequency of the sandwich boom is 220 cycles per minute, while that for the solid laminate is 302 cycles per minute. With the natural frequency well below the minimum operating frequency and with the relatively high damping inherent in reinforced plastic structure, the sandwich type boom offers quite desirable characteristics. With some change of geometry and a probable sacrifice in weight, a similar condition could be accomplished with a solid laminate. However, if the damping is sufficient, it may be unnecessary to avoid exciting frequencies.

Cost analysis of the reinforced plastic configurations is given in Table II.

TABLE II
COST ANALYSIS OF H-23 TAIL BOOM

	Unit Cost	
	Quantity of 10	Quantity of 100
Configuration I	$ 3160	$ 1330
Configuration II	2710	1445
Configuration III	475	395
Configuration IV	2510	850
Configuration V	2355	1150
Federal Stock Catalog "Spares" cost of the existing hardware is:		
H-23 Tail Boom	$ 3560	

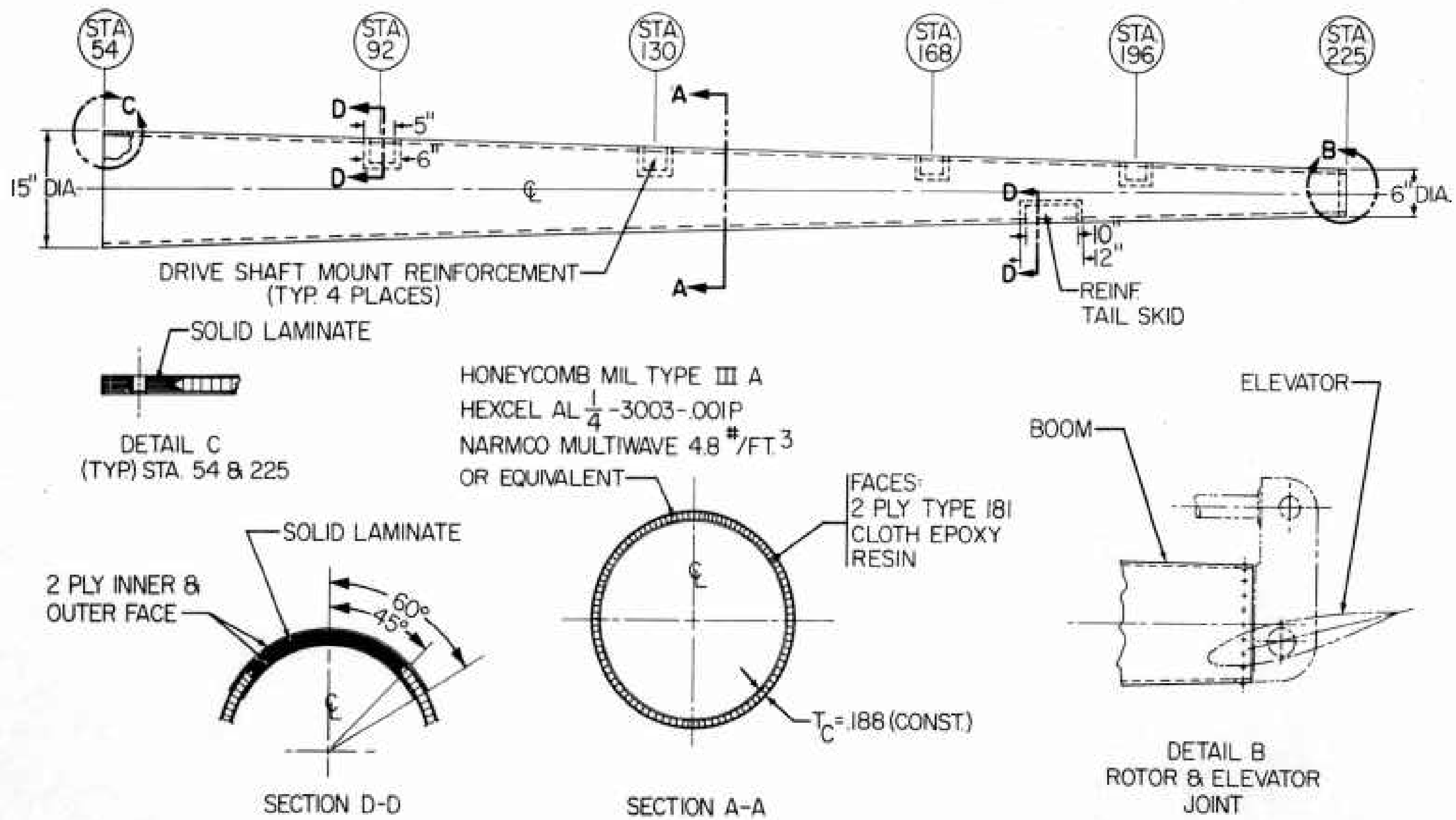


FIGURE 12. H-23 TAIL BOOM OF REINFORCED PLASTIC CONFIGURATION I, ONE PIECE SANDWICH

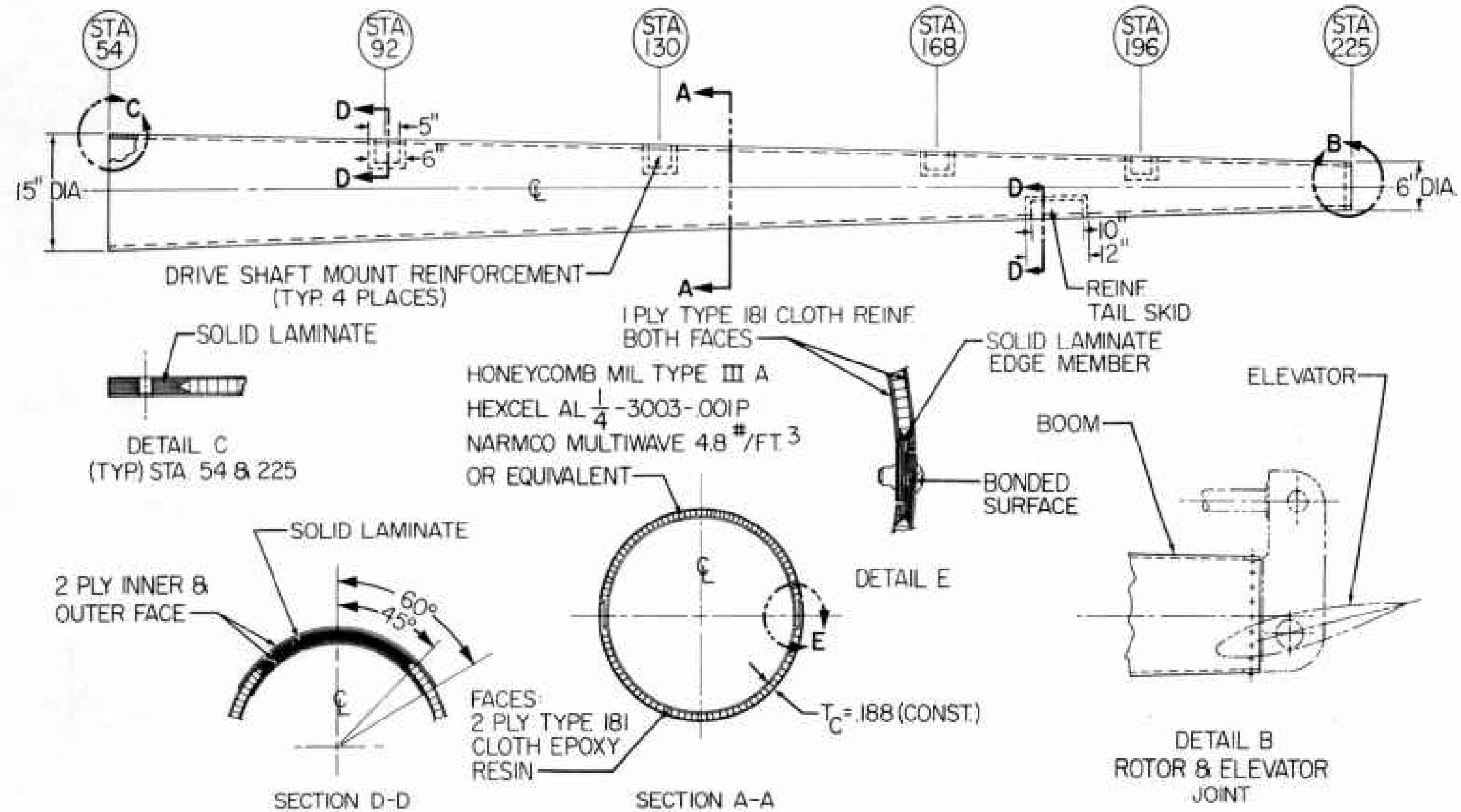


FIGURE 13. H-23 TAIL BOOM OF REINFORCED PLASTIC, CONFIGURATION II, TWO PIECE SANDWICH, SPLIT AT HORIZONTAL CENTERLINE

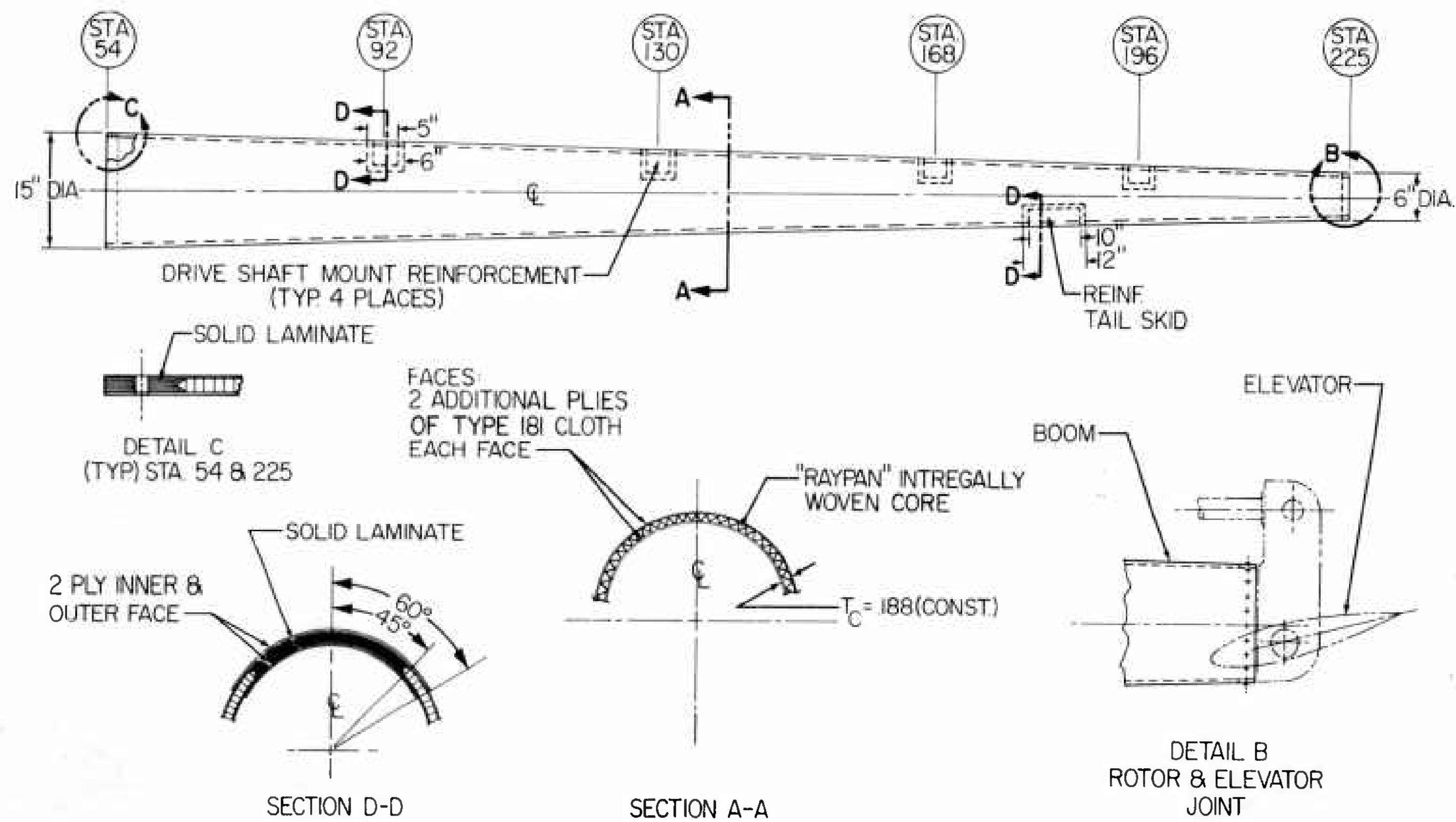


FIGURE 14. H-23 TAIL BOOM OF REINFORCED PLASTIC, CONFIGURATION III, ONE PIECE SANDWICH

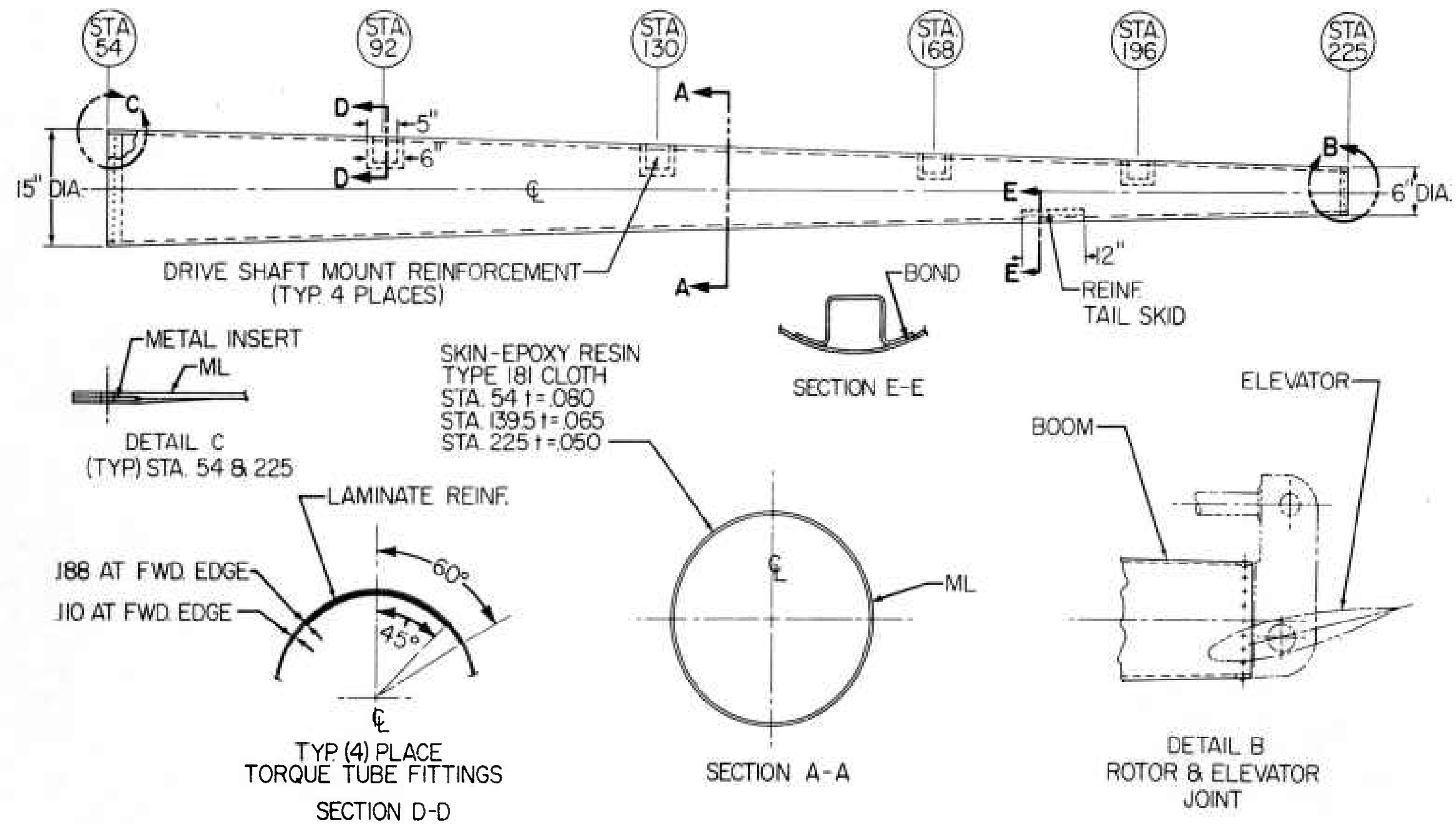


FIGURE 15. H-23 TAIL BOOM OF REINFORCED PLASTIC, CONFIGURATION IV, ONE PIECE SOLID LAMINATE

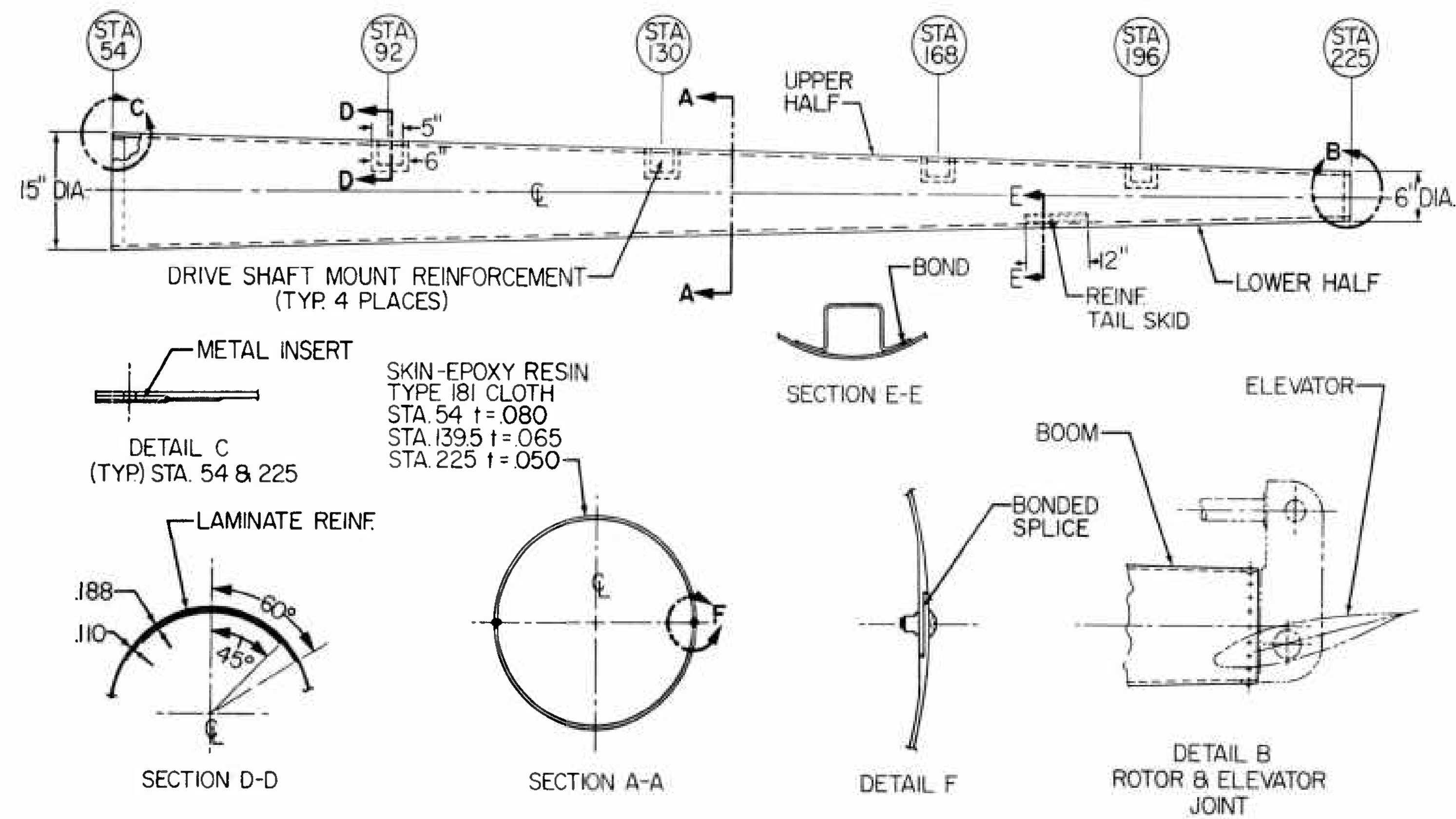


FIGURE 16. H-23 TAIL BOOM OF REINFORCED PLASTIC, CONFIGURATION V, TWO PIECE SOLID LAMINATE, SPLIT AT HORIZONTAL CENTERLINE

METHOD OF MANUFACTURE

The HU-1 and the H-23 tail booms have similar configurations; that is, both incorporate a tapered tubular design. The magnitude of the cross section is greater for the HU-1; however, the manufacturing procedure will be basically the same for both units.

The sandwich configurations for the HU-1 and H-23 tail boom and other body components using this type of structure can be fabricated by a single lay-up and cure, or a three-stage procedure. The three stages will probably result in a sandwich having higher strength. The method is described but is not recommended for these components because of the higher cost relative to the advantages.

The faces are first laid up and precured in separate molds. The sides of the faces to be bonded to the core are then lightly sanded and cleaned. The faces, core, edge members, and local reinforcements are then placed in position. An adhesive is used between the faces and core. The entire assembly is then cured in an autoclave.

Additional work consists of finishing as required, trimming, some machining of mating surfaces for installation, and drilling holes for attachments. The use of an adhesive adds some weight, but this three-stage method results in a much stronger and more reliable sandwich than fabricating the complete sandwich in one curing operation.

The single lay-up and cure method is considered to be the most feasible for these components using sandwich. In this method, the outer face plies, reinforcements, edge members, etc., the core, and the inner face plies are laid up and the complete assembly is cured in an autoclave. This method is less expensive and satisfactory results can be obtained, as evidenced by the design analysis requirements and the test results.

Another method of fabrication that is feasible for this one-piece configuration is filament winding. After the inner face is filament wound, the reinforcements, core, etc., are bonded in place similar to the previous process. The outer face is then filament wound over the core. This method of fabrication has been successfully used for sandwich cylinders. However, previous applications have been used in achieving minimum weight, with strength being a secondary consideration. Strength data are needed to evaluate this technique. It is believed that the method deserves further consideration. It could conceivably be more economical due to automation of the process.

The high strength of filament winding can also be utilized by the use of a new filament orientated preimpregnated material in place of the type 181 cloth as used in this design study. The material was developed by Hercules Powder Co. and is made by winding impregnated glass roving on a cylindrical mandrel in a predetermined helix. The cylindrical structure thus produced

is then slit axially, flattened and molded, or the material is B-staged and used as any other preimpregnated material. The purpose of this material is to combine the preorientation of filaments inherent in filament winding with the shape flexibility inherent in molding flat preimpregnated reinforced plastic. Additional data on this material can be found in the section on filament winding.

The work necessary to form the core is dependent on the type of core used. Glass reinforced plastic honeycomb can be heat formed. Shaping for these applications would probably require a combination of machining and heat forming. The use of reinforced plastic honeycomb core is not warranted unless its special properties such as radar transparency are required. Aluminum honeycomb can be obtained in the unexpanded form, machined to the required taper, expanded and then formed to the required contour. An easily formed core material such as NARMCO Multiwave would reduce the forming time but would increase the machining time if this core is tapered. It is also slightly heavier than the hexagon cell honeycomb. A foam core would require premolding in sections. Of the several potential cores, aluminum honeycomb is the most economical and appears to have the greatest overall advantages.

The one-piece solid laminate construction as shown in Configuration IV, Figure 15, for the H-23 helicopter may be fabricated by two basic methods, with each method utilizing a male mandrel. A woven fabric is laid up complete with local reinforcements and is cured in an autoclave, or the complete boom can be filament wound. This type of component is ideally suitable for filament winding. These methods of solid fabrications are by far the most economical.

The two-piece constructions, as shown on Figures 7, 8, 13, and 16, are fabricated utilizing an open-face mold. The methods of construction are identical to the sandwich and solid lay-up procedures as outlined for the one-piece article. However, since this method utilizes an open-face mold, a vacuum bag or pressure bag, as well as the autoclave, may be used for the required pressure to assure a structurally sound item.

Additional testing was required to verify some of the material characteristics used in this study. Bending, compression and panel shear tests of solid laminates and sandwich panels were made. Details of these tests are included in another section of this report. The development of a full-scale tail boom will require considerably more testing beyond the scope of this program to determine the optimum design and fabrication technique.

The laboratory tests that were accomplished in this program indicate that the necessary strength as determined by the analytical design studies can be obtained.

Considerable difficulty was experienced in bonding of the preimpregnated polyester faces of all types of core material. A separate adhesive was used and is necessary. Satisfactory bonds can be obtained using polyester, but compatibility of materials and the cure cycle must be investigated.

Higher strength and a more consistent sandwich can be obtained with epoxy resin than with polyester. Epoxy resin is an excellent adhesive and the necessity of a separate adhesive is eliminated. The cost should therefore be lower. It is recommended that aircraft body structures be fabricated using epoxy resin.

From consideration of strength alone, the plastic design can be comparable in weight and in many cases lighter than the metal design. Generally speaking, for comparable strength designs, the plastic structure will be more flexible than metal. Statically, this difference in flexibility is considered unimportant in most cases. Dynamically, the relative merits depend upon the particular conditions, and further study is required.

From consideration of fatigue, the combination of 181 cloth and epoxy resin appears to have an advantage over the more common aluminum alloys. Based on existing data, if one plots percentage of ultimate stress against number of cycles to failure, the curve for this reinforced plastic will be above that for aluminum. In addition to this, the normal fabrication techniques for reinforced plastics provide a much more continuous structure; i.e., fewer holes for attachments, discontinuities, etc., than is possible in aluminum. Where discontinuities do occur in plastic, it is easier to provide compensating reinforcement. This characteristic of plastic design minimizes stress concentration, which is the primary cause of fatigue failures.

Reinforced plastics offer a greater degree of internal damping than do metals. However, there are no data available to permit a quantitative evaluation of this characteristic. Appreciable testing, both specimen and full scale, is required to determine the actual significance of this increased damping.

EVALUATION

The configurations which have been studied in detail represent the most promising of all the potential design and fabrication approaches which have been investigated. In all cases, feasibility is indicated. Detailed analysis has been accomplished only in areas pertinent to a preliminary evaluation of feasibility. Optimization of the reinforced plastic designs to the degree that the comparable existing metal designs have been optimized is not intended or justified.

HU-1A Configuration II, sandwich construction split along the vertical centerline, indicates lowest cost, the quantity production cost being

appreciably less than indicated cost of the conventional metal boom. The integral boom and vertical fin seem to be the most significant factors in the lower fabrication cost achieved in this configuration.

All reinforced plastic H-23 configurations show very significantly lower cost than the existing metal boom. An inconsistency apparently exists between the Federal Catalog prices for the HU-1A and H-23.

Under any circumstances, it appears that reinforced plastic tail booms are very promising applications from a cost standpoint.

Advantages

1. Generally better strength to weight ratio.
2. Better fatigue characteristics.
3. Superior damping qualities.
4. Can proof test to higher loads without damage to structure.
5. Greater durability.
6. Probable lower cost.

Disadvantages

1. Less design and fabrication knowledge available.
2. More difficult process control.
3. Less reliable inspection techniques.

RECOMMENDATIONS

It is recommended that development of a typical or specific reinforced plastic tail boom be accomplished in accordance with the following sequence:

1. Perform a complete design and analysis for a reinforced plastic tail boom for a specific application supplemented by laboratory testing as required.

2. Fabricate one or more test articles maintaining detailed records for cost analysis.

3. Perform static and dynamic tests on the completed article.

4. Make evaluations and design changes as indicated by tests.

5. Fabricate and install tail boom on a flight vehicle for evaluation.

6. Make a comparative analysis of the reinforced plastic and metal boom designs.

LANDING GEAR DESIGN STUDY

The landing gear on most Army aircraft employ an oleo or a cantilever beam spring shock absorber. The cantilever beam spring is particularly suitable for utilizing the unique properties of glass reinforced plastic. This type is used on fixed wing aircraft and on helicopters. This study is concentrated on this type and considers two basic cantilever beam spring landing gear struts. These are the helicopter skid gear and the light fixed wing aircraft fixed strut wheel type gear. These two landing gears are considered separately because of the different approach in applying the spring principle in the design of the gear.

Glass reinforced plastics are excellent energy absorbers. They dissipate energy faster than metals; therefore, vibrations damp out quickly and smoothly. The energy absorbing capacity of unidirectionally reinforced plastic is more than twice that of steel. The high energy absorbing ability of these materials results from their high usable strength and low Young's modulus.

The use of reinforced plastics in energy absorbing applications, such as flat springs in industrial machinery, is becoming significant. Kaman Aircraft Corporation has fabricated a rear spring for the Chevy II automobile from unidirectional fabric with a 60 percent weight reduction over the existing steel spring. Very few applications for aircraft landing gear are known. Malmo Flygindustri of Sweden has developed a cantilevered filament wound fiberglass landing gear strut for their MFI-10 STOL aircraft (Reference 1). Full scale tests have been very successful. The fiberglass strut gives a smoother ride than the regular gear and induces noise. The weight is somewhat lighter than the metal gear. A new prototype aircraft for commercial use built by Bede Aircraft uses a landing gear strut fabricated from unidirectional glass fabric. It is also reported that Piper Aircraft is using a fiberglass strut on their all-plastic aircraft currently under development, but no information is available.

Some work has been accomplished in Germany on two applications of reinforced plastics in landing gears and is reported in Reference 73. One type considers the use of fiberglass wound case for a conventional oleo strut. Figure 17 shows a comparison of steel and fiberglass reinforced plastic as liquid spring case material. Fiberglass reinforced plastics appear to have considerable advantage over steel in this application.

A second type of plastic spring for a light aircraft landing gear proposed by Hanle (Reference 73) uses filament wound rings loaded in tension. The spring consists of a series of double conical inner rings of steel and outer rings consisting of middle-hard polyamide bodies on which the resin impregnated glass rovings were wound under prestress, Figure 18. This type of spring uses the reinforced plastic most efficiently in tension. When a compressive load is applied to the spring, the steel ring is forced into the polyamide ring. The filament wound ring carries the load primarily in hoop tension and absorbs energy in expanding. The reference states that a strut using this type of spring has been used successfully on a light aircraft.

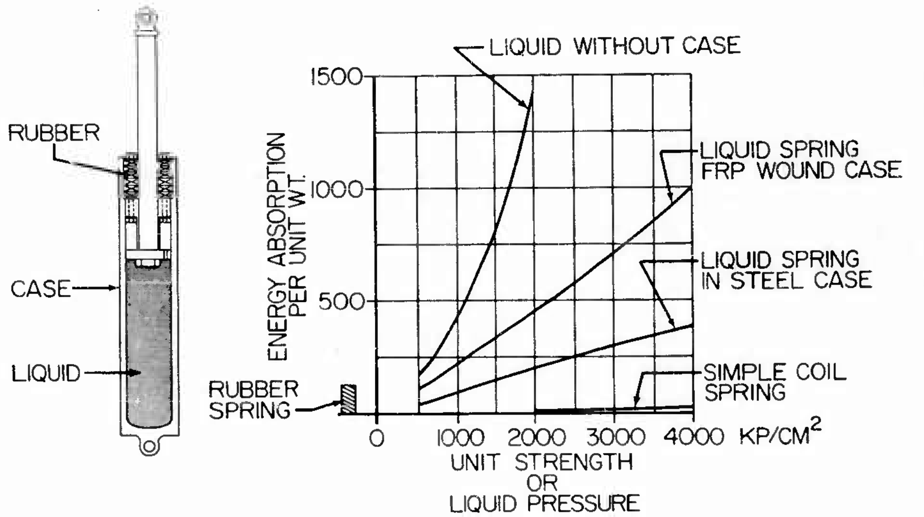


FIGURE 17. COMPARISON OF STEEL AND FIBERGLASS REINFORCED PLASTIC AS A LIQUID SPRING CASE MATERIAL

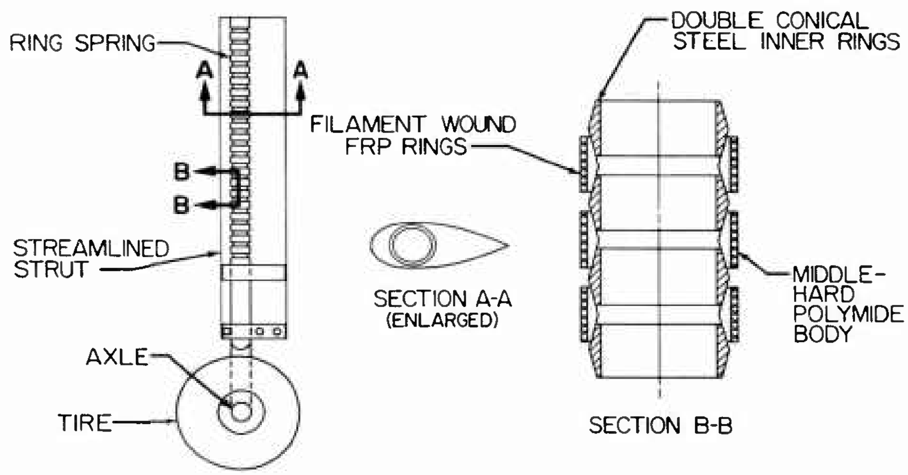


FIGURE 18. RING SPRING SHOCK STRUT OF REINFORCED PLASTIC

HELICOPTER SKID LANDING GEAR

The skid type landing gear is used on many Army helicopters of the light observation and utility classes and is to be used on the LOH aircraft currently under development. In the simplest form, it is composed of two primary elements--the skids and two energy absorbing cross tubes. The cross tubes are attached to the fuselage structure, usually at four attach points, in such a way that they are not restrained in torsion. Loads Loads which would cause torsion if the cross tubes were so constrained are taken out as bending mements in the skids and cross tubes; hence, the rigidity of this type of skid gear is not dependent upon the fuselage structure. This study is confined to the use of reinforced plastics in the shock absorbing components of this type gear.

The design criteria for skid gear are relatively meager; however, general landing gear criteria which are considered applicable to this study are contained in Reference 7, ARDC Manual 80-1, Reference 3, ANC-2 Bulletin, Reference 50, MIL-S-8698, and Reference 70, WADC TR 58-336.

The requirements specify that the gear shall not yield when subjected to limit landing loads nor fail when subjected to ultimate landing loads (Reference 50). The requirement pertaining to the limit landing conditions is not generally complied with in practice. Current design practice is to permit the cross tubes to yield well below the limit landing loads, hence absorbing a large portion of the landing energy by plastic deformation. This design approach is based on the supposition that replacement of badly deformed cross tubes is acceptable to the user in lieu of the decreased helicopter performance associated with heavier "elastic" gear. The practice has been approved by the CAA for civilian helicopters (Reference 70). However, no military specification has specifically approved the practice to date. The "yielding" type gear has apparently been fairly satisfactory in practice albeit it has disadvantages.

The ultimate strength of helicopter landing gears is specified by two requirements in Reference 50, as follows:

1. The structure shall support, without failure, ultimate loads resulting from loading conditions incorporating an ultimate factor of safety of 1.5.

2. During the reserve energy drop test demonstration, failure of the structure shall not occur at a vertical descent velocity equal to the limit vertical descent velocity times the square root of 1.5.

Requirement (1) above is specified as a factor of safety for the entire aircraft and, therefore, would ordinarily be interpreted as a requirement for the landing gear. Requirement (2) is less critical and is an ultimate requirement for the landing gear in particular. However, since it is a demonstration test requirement, it does not necessarily conflict with the 1.5 design factor. Nevertheless, authoritative interpretation indicates

that the factor of safety of 1.5 does not apply to the landing gear mechanism and that the reserve energy requirement is appropriate for design (Reference 70).

In order to provide a basis for comparative evaluation of reinforced plastics and conventional design, two existing aircraft configurations were selected for study. The Bell Model 47, which is similar to the Army H-13, represents the smaller aircraft in the light observation class. The Model 47 was selected because more detailed information was available. The HU-1 represents the larger utility class helicopters. These landing gears are constructed of round tubing for the shock absorbing members and skids.

The primary stresses in the cross members result from longitudinal loads; therefore, the reinforcement fiber orientation should be longitudinal in order to develop maximum bending strength to resist the applied loads. This can be accomplished by filament winding. The feasibility of this process is dependent upon the geometry of the component. This analysis is based on the use of a unidirectional fabric such as "Scothply", manufactured by Minnesota Mining and Manufacturing Company. This is a unidirectional nonwoven fabric preimpregnated with epoxy resin and is considered to be representative of the available high-strength materials. New materials having higher strength and moduli are under development and can be used to optimize future designs.

The mechanical properties of "Scotchply" Type 1002 for a stress angle of 0^{o} as given in Reference 54 are summarized in Table 3.

TABLE 3
MECHANICAL PROPERTIES OF "SCOTCHPLY" TYPE 1002

Property	Dry - 70^{o}F* (lbs/in^2)	Dry - 160^{o}F* (lbs/in^2)
F_t	110,000	102,000
F_c	80,000	57,500
F_b	130,000	106,000
E	5.5×10^6	-

* The wet strength retention factor = .86

The structural design criteria used in this investigation are summarized in Table 5. These criteria are based on Reference 50. As shown in Table 5, the design criteria used for the Bell Model 47, Reference 14, deviate from Reference 50. At the risk of complicating the comparison of the reinforced plastic and conventional gear, the design criteria of Reference 50 are adhered to in this study. An investigation of the criteria specified in MIL-S-8698 is reported in Reference 70. This report indicates that the requirements may be conservative. However, there are no known military authorized deviations.

Two basic methods of reinforced plastic construction were investigated for the landing gear shock absorbing components for both the Bell Model H-47 and the HU-1. These were solid laminates and sandwich construction. The method of analysis for each is similar. The designs investigated are shown in Figures 19, 20, 21 and 22.

Laminates for these components should be molded at moderate to high pressure. Matched metal molds are the most desirable if the quantity justifies the cost. For experimental and low quantity production, a female metal mold and pressure bags could be used. Some experimentation would be necessary to develop the molding technique and cure cycles for the thick sections. It may require lay-up and cure by successive steps; however, Kaman, in fabricating the automobile spring, had no trouble molding the thick section in one step.

The faces of the sandwich configuration should be premolded and bonded to the core by a separate operation. The local reinforcements required for attachments would also be premolded parts.

Additional strength and durability can be obtained by wrapping both types of struts with a single layer of woven cloth after other operations are completed.

The cost analysis for the shock absorbing members for the various reinforced plastic configurations is summarized in Table 4.

TABLE 4
COST COMPARISON OF HELICOPTER LANDING GEAR

Configuration	Unit Cost Quantity of 10	Unit Cost Quantity of 100
Model 47, Solid Laminate	$1070	$615
Model 47, Sandwich	980	570
HU-1A, Solid Laminate	1105	650
HU-1A, Sandwich	1015	550

For approximate comparison, the spares cost of existing components as listed in the Federal Stock Catalog are as follows:

H-23	$215
H-13	95
HU-1A	105

This apparent cost disadvantage for reinforced plastics cannot be taken at face value. The cost of the reinforced plastic design is considered to be conservative because of the necessary development required to optimize the first product. The costs of the present spares as listed in the Federal Stock Catalog are believed to be unrealistically low.

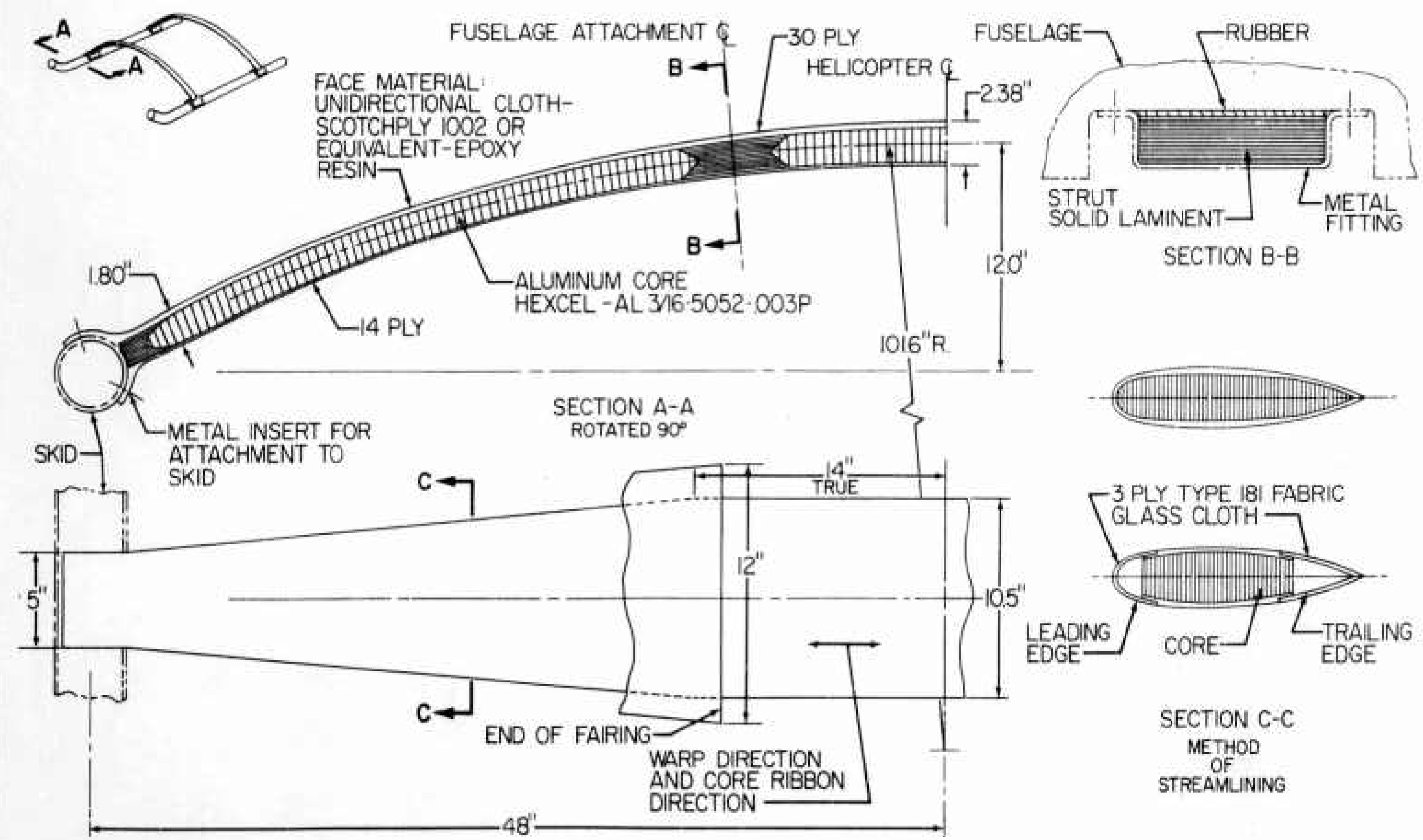


FIGURE 19. HU-1 LANDING GEAR OF REINFORCED PLASTIC SANDWICH CONSTRUCTION

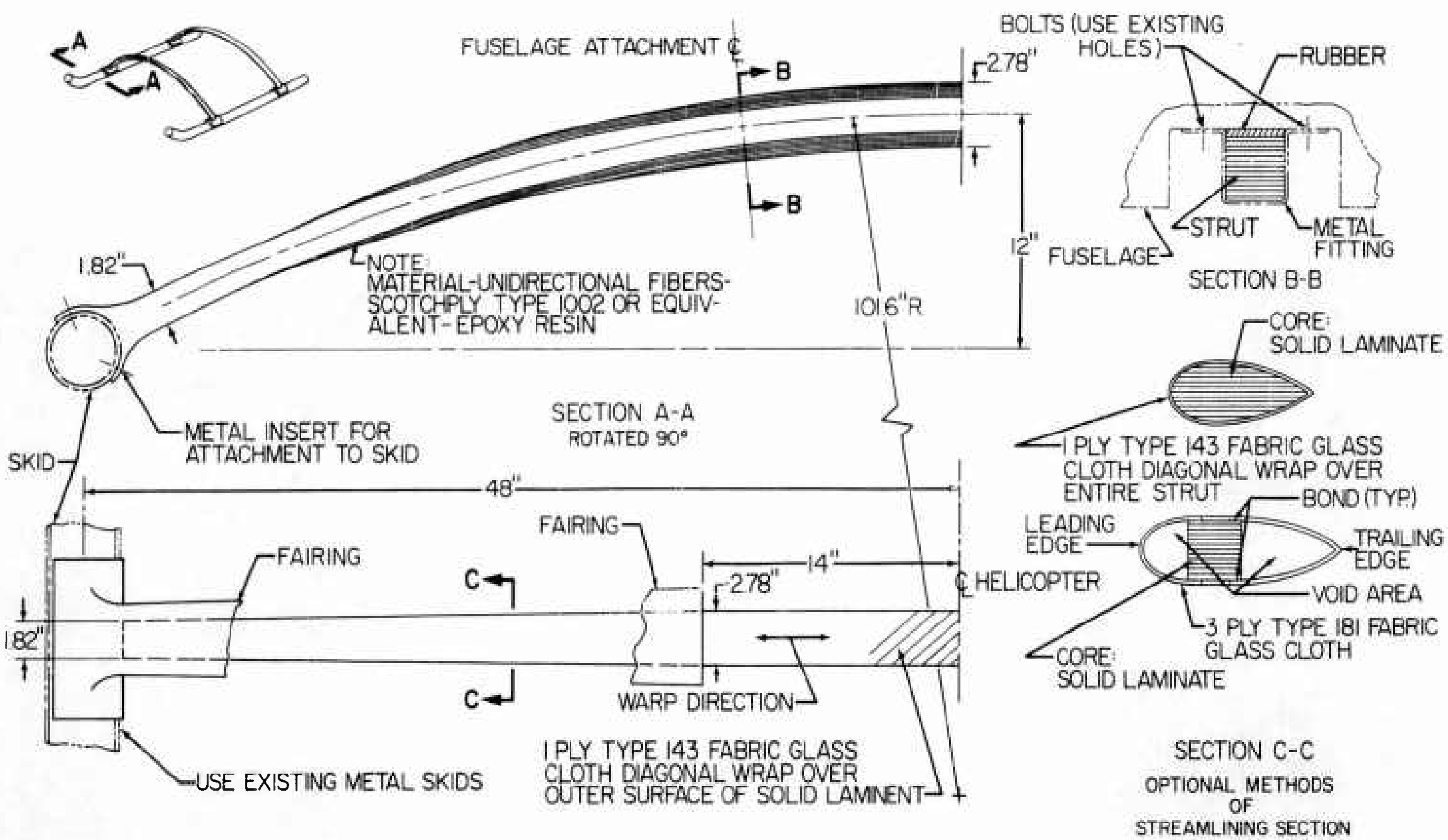


FIGURE 20. HU-1 LANDING GEAR OF REINFORCED PLASTIC, SOLID LAMINATE CONSTRUCTION

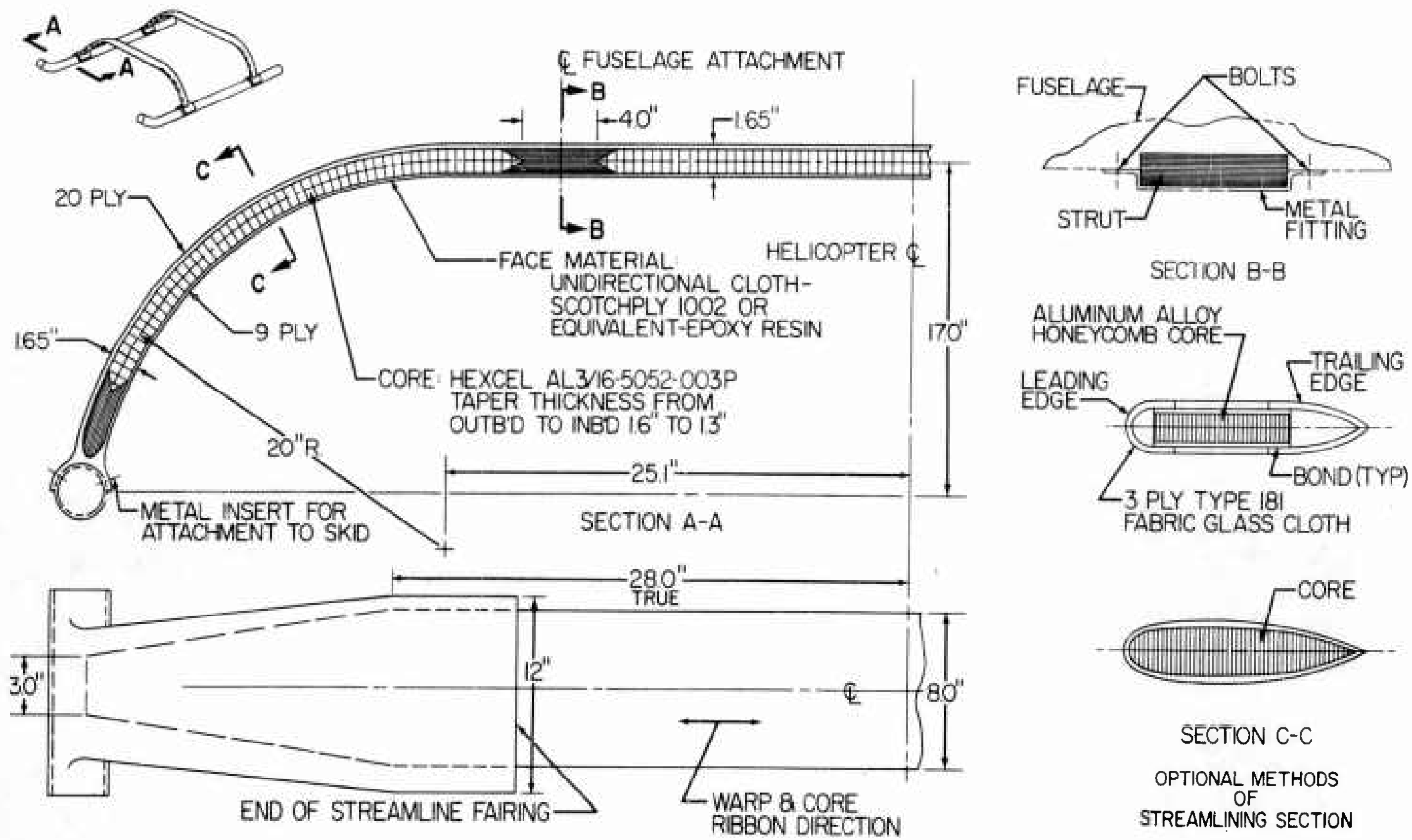


FIGURE 21. BELL MODEL 47 LANDING GEAR OF REINFORCED PLASTIC SANDWICH

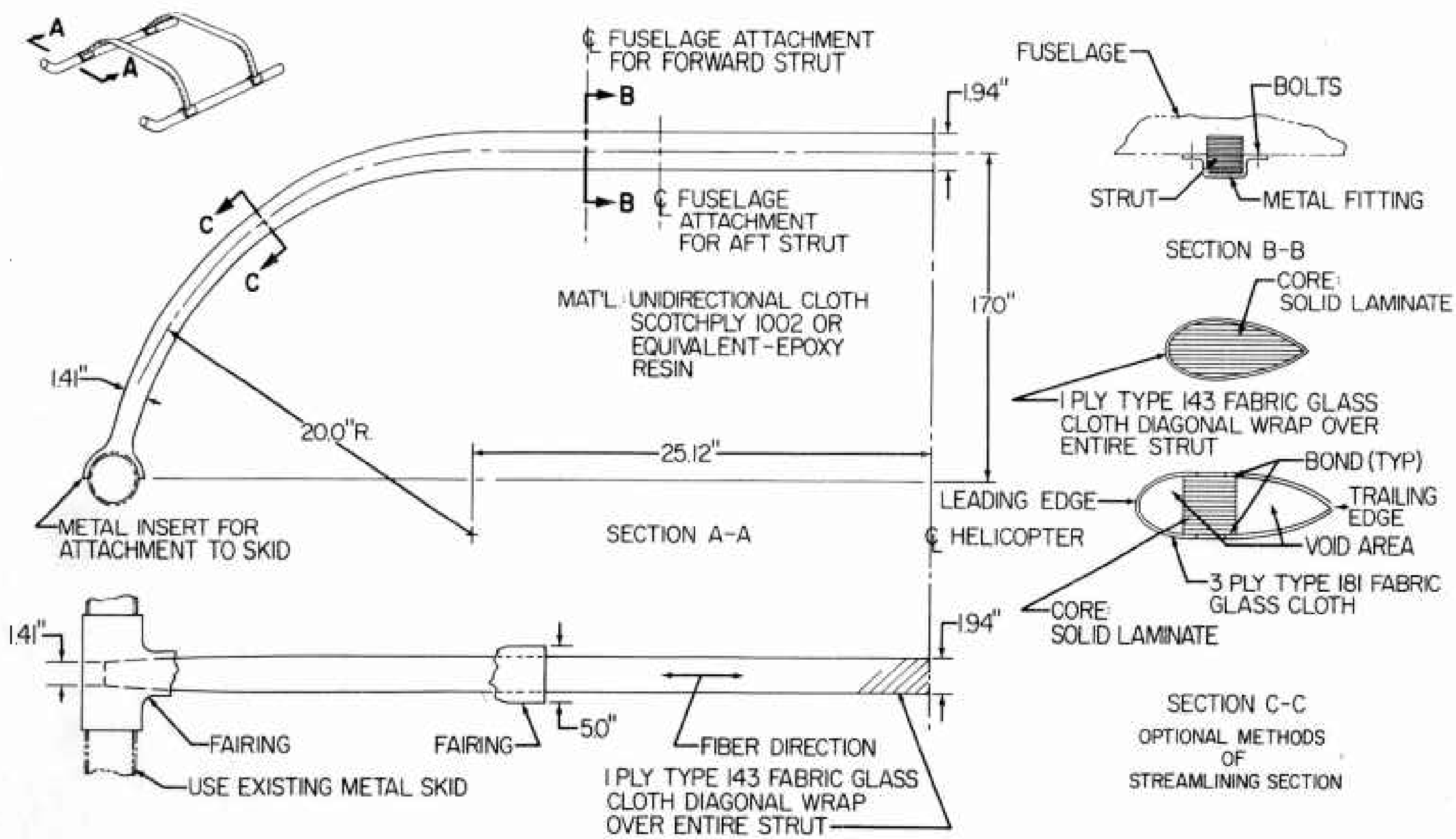


FIGURE 22. BELL MODEL 47 LANDING GEAR OF LAMINATED REINFORCED PLASTIC

Evaluation

The most popular helicopter skid gear currently in use is the "yielding" type constructed of aluminum tubing. The "yielding" gear absorbs a large portion of the ultimate landing condition energy by plastic deformation (approximately 80 percent of the total energy). The "yielding" gear has the following advantages over eleastic gear constructed of aluminum tubing:

1. The gear is much lighter, hence incurring less weight penalty and improving helicopter performance.

2. The landing loads are reduced appreciably, thus affording greater protection to the aircraft structure and the occupants.

However, this type of gear also had disadvantages:

1. "Yielding" gear design requires the replacement of the energy absorbing portion of the gear after "hard" landings, due to permanent deformation.

2. The majority of tubular constructed landing gears contain numerous intersections resulting in inordinate parasite drag.

To state that fiberglass reinforced plastic skid gear retains the advantages of "yielding" tubular aluminum gear while eliminating the disadvantages would be an oversimplification; however, the statement is approximately true.

The design criteria, deflections, maximum load factors, and weights of the present metal gear and the reinforced plastic designs are summarized in Table 5.

The requirements of Reference 50 are generally found to be conservative as pointed out by Reference 70. Actual measurements on existing utility helicopters show that the maximum sinking speed is usually less than 5 feet per second and that rotor lift varies from 80 to 90 percent of the normal gross weight (Reference 70). Figure 23 shows that the landing gear load factor for sinking speeds in the range of 0 to 6 feet per second is appreciably lower for reinforced plastic gear compared to "yielding" aluminum gear. At higher sinking speeds, the reinforced plastic gear, being elastic, will develop aircraft load factors that exceed the maximum landing gear load factor of the aluminum gear; however, the aircraft structural ultimate load factor is not exceeded. This really constitutes an advantage for the reinforced plastic gear since the greatest damage to aircraft structure and equipment is attributed to repeated loads encountered in normal operation (which are lower with reinforced plastic gear) than upon an occasional high loading (Reference 70). The maximum ultimate load factor developed by the reinforced plastic gears is greater than for the yielding metal gear (Table 5), but is still less than the structural design load factor. Although the load factors are slightly higher, the energy from a "hard" landing can be absorbed without failure of the landing gear components.

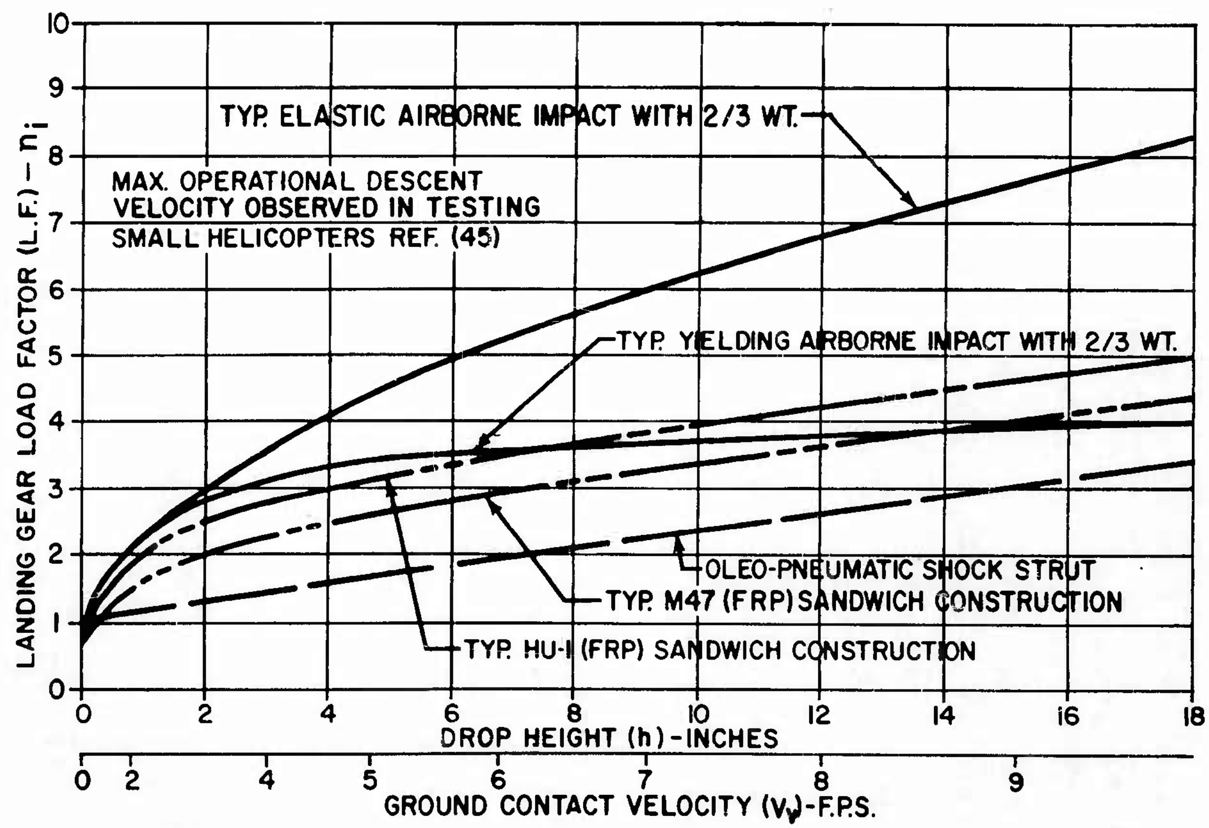


FIGURE 23. COMPARISON OF IMPACT LOAD FACTORS FOR CONVENTIONAL AND REINFORCED PLASTIC HELICOPTER SKID LANDING GEAR

TABLE 5
HELICOPTER SKID LANDING GEAR - SUMMARY

Item	MIL-S-8698 Requirements	Bell Model 47				Bell Model HU-1		
		Existing Design	Metal Tube to MIL-S-8698	Fiberglass Solid Laminate	Fiberglass Sandwich Construction	Existing Design	Fiberglass Solid Laminate	Fiberglass Sandwich Construction
Design Limit Sinking Speed (FPS)	8.0	7.5	8.0	8.0	8.0	Unknown	8.0	8.0
Design Ultimate Sinking Speed (FPS)	9.81	9.17	9.81	9.81	9.81	Unknown	9.81	9.81
Design Rotor Lift At Impact (Lb)	.67W	.80W	.67W	.67W	.67W	Unknown	.67W	.67W
Weight of Landing Gear Shock Absorbing Members (Lb)	-	40.0	40.25	46.0	48.0	47.2 (Est.)	79.6	86.0
Maximum Cross Member Deflection (In.)	-	8.5	8.5	13.0	15.0	10.5 (Est.)	9.7	11.3
Ultimate Landing Gear Load Factor (n_j)	-	3.85	4.1	4.87	4.35	3.44	5.05	4.93
Ultimate Helicopter Load Factor (n)	-	3.6	3.93	4.29	3.80	3.30	4.33	4.29

The design philosophy of metal skid gear assumes that replacement of the energy absorbing gross members is acceptable to the helicopter user. Replacement of a badly deformed cross member in Army field operation may not be practical. At any rate, this constitutes a logistics problem and an additional item for inspection and maintenance. The reinforced plastic gear incurring only elastic deformations will ordinarily not require replacement.

The HU-1 skid gear accounts for approximately 32 percent of the total parasite drag (Reference 14). On a cleaned-up helicopter design analyzed in Reference 23, the conventional skid gear accounted for 50 percent of the total parasite drag, as shown in Figure 24. This reference showed that the gear parasite drag could be reduced to one-third the original value by:

1. Providing a streamlined section.

2. Reducing the number of intersections (making the cross members in one piece).

3. Designing all remaining intersections to intersect at right angles (eliminating oblique angles).

All these requirements are compatible with existing reinforced plastic fabrication techniques; hence, reinforced plastic design offers a convenient way of improving aerodynamic efficiency of advanced helicopter designs. For high-performance VTOL aircraft employing skid type gear, these improvements are almost mandatory.

The weights of the reinforced plastic struts are considered to be competitive with the metal designs. Mechanical properties used in the study are considered to be conservative. Although the weight computed for the fiberglass design for the HU-1 is greater than the metal components, it is believed that this weight disadvantage can be removed with an optimized design based on less conservative mechanical properties and a test evaluation. Newer high-strength glass reinforced plastics presently under development will reduce the weight significantly.

The advantages and disadvantages of reinforced plastic helicopter skid landing gears are summarized:

<u>Advantages</u>

1. A glass reinforced landing gear has a higher energy absorbing ability than steel and aluminum alloys commonly used for the shock absorbing components.

2. The landing load factors are lower for normal rates of descent.

3. There is less wear and tear on aircraft structure and equipment because of lower loads.

4. It is more comfortable for aircraft occupants.

5. It is capable of reacting loads from "hard" landings without failure.

6. The necessity of replacing the energy absorbing member after a "hard" landing is eliminated. (A metal member that yields due to excessive load required replacement).

7. It improves helicopter performance through less drag. Fabrication methods are adaptable to developing streamlined components with fewer intersections.

8. There is potential lower <u>overall</u> cost by eliminating the replacement of yielding type metal parts.

<u>Disadvantages</u>

1. Potential higher weight - the weight differential shown in this analysis can be substantially reduced or eliminated by careful design and fabrication optimization.

2. Higher initial cost.

<u>Conclusions</u>

1. Glass reinforced plastics are ideally suited to energy absorbing applications such as helicopter skid landing gears.

2. Design and fabrication of reinforced plastic landing gears are feasible within the current state of the art.

3. Reinforced plastic skid type gears are competitive weight-wise and cost-wise with conventional metal gears - assuming further optimization of the plastic design and processes.

4. Many advantages accrue to the reinforced plastic gear as noted in the "Evaluation" with no significant disadvantages.

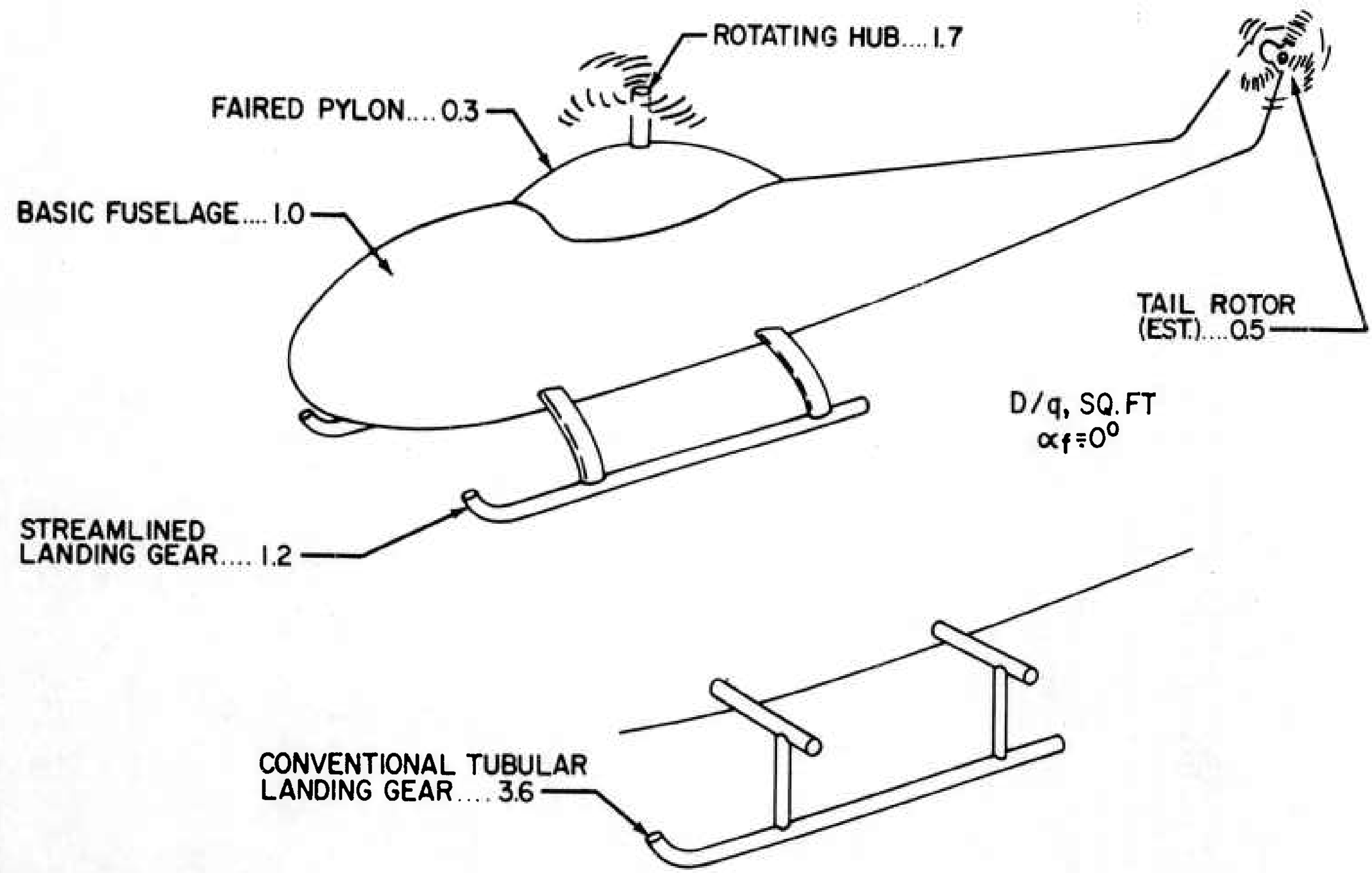


FIGURE 24. EQUIVALENT PARASITE DRAG AREA OF VARIOUS HELICOPTER COMPONENTS

L-19 LANDING GEAR STRUT

The landing gear of the Army L-19 aircraft was chosen for study because it is considered to be representative of current and contemplated light fixed wing Army aircraft using a nonretractable landing gear. Conclusions resulting from a study of the feasibility of using reinforced plastics for the shock absorbing components of the L-19 landing gear will also be applicable to helicopters using similar type gears.

A glass reinforced plastic strut for an aircraft similar in size and weight to the L-19 has been successfully drop and flight tested by Malmo Flygindustri of Sweden (Reference 1). This strut is based on their MFI-10 Vipan aircraft weighing approximately 2590 pounds. The weight of the L-19 is 2400 pounds. Glass roving reinforced polyester resin was used for the first prototype. Epoxy will be used for future components. The manufacturer reports that highly satisfactory performance has resulted from tests. He also reports that landing and taxiing load factors are reduced, resulting in a smoother ride with less vibration transmitted to the aircraft.

The main landing gear strut used on the L-19 aircraft is typical of the cantilever beam spring method of absorbing landing shock incorporated on some Army aircraft. Each main gear strut (one left-hand and one right-hand) is a single piece of chrome-vanadium steel heat-treated to 240,000 psi ultimate tensile strength. The strut is 0.7 inch thick throughout its length; the width tapers from 6.0 inches at the upper end to 1.5 inches at the axle. The upper end of each strut is bolted to the lower portion of the fuselage landing gear bulkhead assembly. A cantilevered axle is bolted directly to the lower end of the strut.

U. S. Government Bulletin ANC-2, "Ground Loads", Reference 3, establishes the minimum structural design requirements for all aircraft. A review of this document indicates that the "Two Wheel Level Landing Condition" is probably the critical condition for design of the L-19 landing gear. Bulletin ANC-2 specifies that the vertical reactions at the ground shall be those resulting from the design landing speeds and sinking speeds. Since these values are not known for this aircraft, an aircraft load factor of 2.6 (applied) is chosen as the maximum load factor required. This choice is based on design experience and the computed strength of the existing gear. With the load factor known, the maximum ground vertical reaction may be determined from Civil Air Regulations, Part 3, Paragraph 3.234 (Reference 21).

$$P = \frac{1}{2} W_G \left(n - \frac{L}{W_G}\right)$$

where: p = ground vertical reaction

W_G = aircraft gross weight = 2400 pounds

L = wing lift assumed acting during landing, not to exceed

$2/3\ W_G$ = 1600 pounds

n = aircraft load factor

substituting:

$$p = 1200\,n - 800$$

and for a load factor of 2.6,

$$p = 2320 \text{ pounds limit or } 3480 \text{ pounds ultimate.}$$

In order to make a direct comparison of the performance which can be expected from a reinforced plastic landing gear strut versus the existing steel strut, an approximate design for a reinforced plastic strut has been worked out. This is not intended to be a finished, optimum design, but is developed only far enough to illustrate the advantages and disadvantages.

In designing the reinforced plastic landing gear strut, the basic geometry of the existing gear is retained. The reinforced plastic struts attach to the original fitting at the fuselage, and the wheels are in the same position for both configurations when the aircraft is resting on the ground. In addition, provisons are made for bolting the original axle assembly to the new strut. These requirements fix the end positions of the strut. A parabolic curve is selected as the shape of the new strut rather than the straight line form used on the steel. This is done to avoid the sudden change in slope at each end, which is undesirable in reinforced plastics. See Figure 25.

The cross section of the reinforced plastic strut is arbitrarily established as a rectangle with the width tapering from 6.0 inches to 2.0 inches. The thickness used is that thickness necessary to provide the required strength under the design loads. The basic $F_B = MC/I$ formula is used in computing the thickness required. Substituting $\frac{bt^3}{12}$ for I and $\frac{t}{2}$ for C, the formula becomes;

$$\text{thickness} = t = \sqrt{\frac{6M}{F_B b}}$$

where F_B is the desired maximum working stress, taken as 85,000 psi in this case. This is a conservative value for design allowable strength of glass reinforced plastics. It can be easily obtained with present unidirectional fabric or roving laminated with epoxy resins.

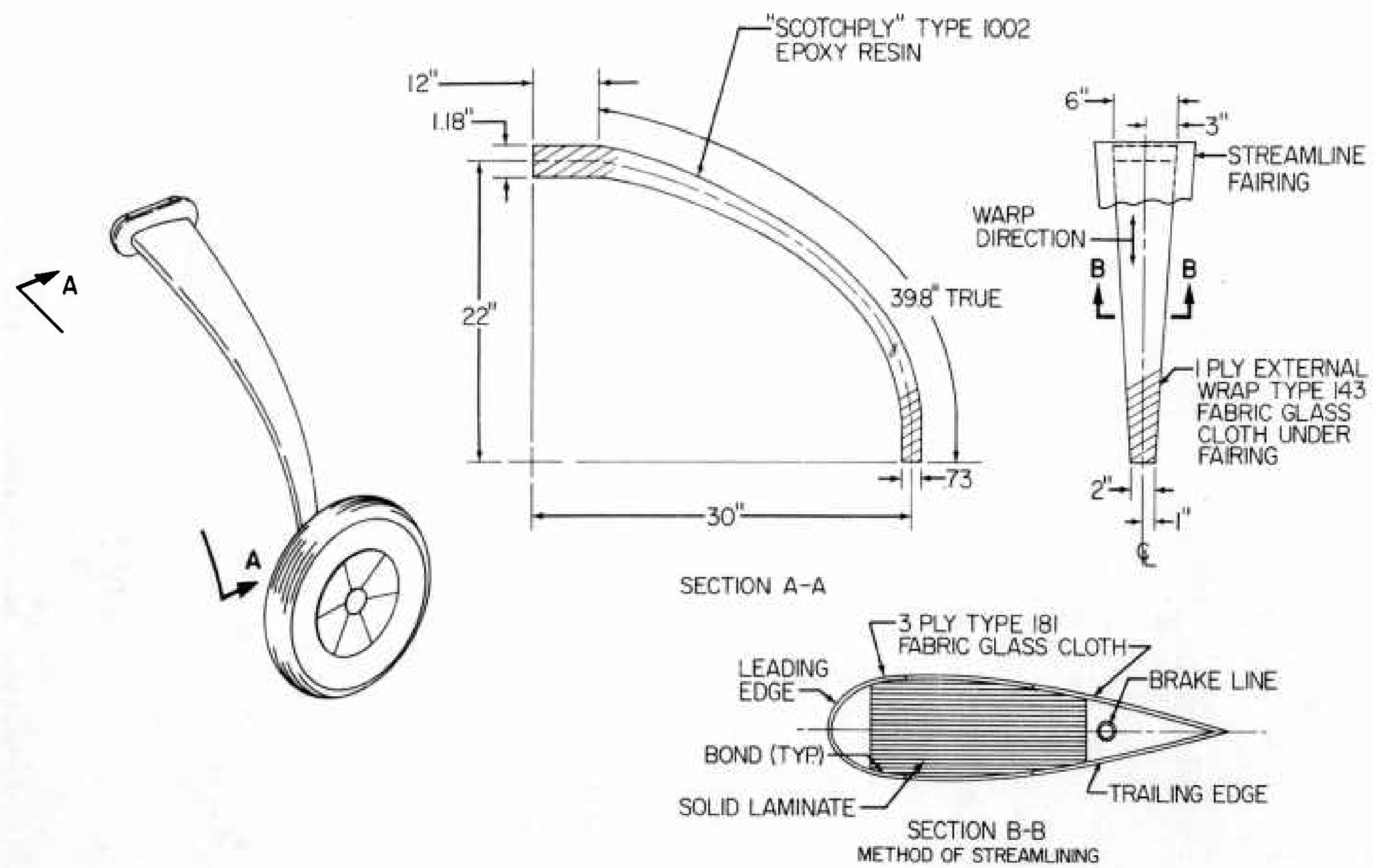


FIGURE 25. L-19 LANDING GEAR STRUT OF LAMINATED
REINFORCED PLASTIC

Then $t = \frac{M}{14150b}^{\frac{1}{2}}$

The primary function of the landing gear obviously is to react the landing and other ground loads with the least amount of shock transferred to the airplane structure. The relative shock absorbing ability of two landing gear struts made from different materials may be illustrated by curves depicting the airplane load factor, n, versus the sinking speed, V_V, of the aircraft. Data for these curves are computed below.

As a first step in determining the performance of the struts, deflections are computed. The Moment Area Method is used to compute these deflections. The M/EI values are computed for the design loading and are plotted as Figure 26.

The deflection at the end of each strut is then determined by integrating the area under the M/EI curve. Results are as follows:

> Deflection at end of steel strut at design load = 8.65 inches.
> Deflection at end of reinforced plastic strut at design load = 14.97 inches.

The curve of Figure 27, showing deflection versus load factor, is then plotted. Deflection is assumed to be proportional to the ground reaction P, applied at the end of the strut. It should be noted that the simplifying assumption is made that the moment arm on the beam is constant throughout the loading range. This is not true because the end of the strut will move in an arc, but the effect will be similar on both struts and is, therefore, ignored in this study. The ground reaction, P, is related to airplane load factor, n, by the previously stated formula, P = 1200n - 800.

The relationship between the deflection of the strut and the sinking speed, V_V, of the aircraft may be established by equating the kinetic energy of the aircraft as it contacts the ground to the potential energy of the deflected strut.

$$\frac{1}{2} \frac{W_G}{g} V + \frac{1}{2} (W_G - L)d = \frac{Pd}{2}$$

Substituting W_G = 2400 pounds; g = 32.2 ft/sec.; $L = \frac{2W_G}{3} = 1600$,

the energy becomes: $18.63V^2 = \frac{Pd}{2} - 400d$

For given values of load factor, n, the velocities, V_V, are obtained using the previously computed relationship between n, p, and d. These values are plotted on Figure 28.

Estimates of the weights of the reinforced plastic landing gear strut and of the steel strut show a marked advantage for the reinforced plastic.

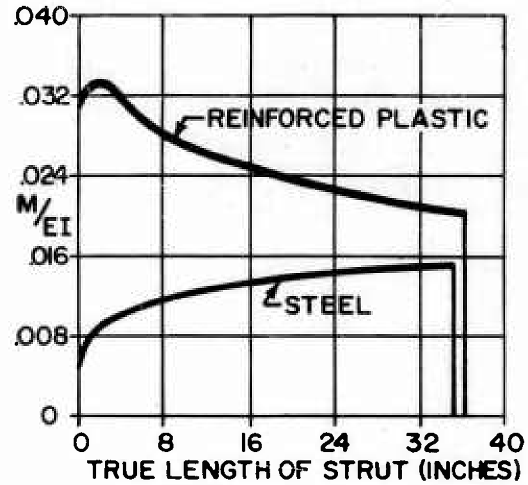


FIGURE 26. M/E1 CURVES FOR L-19 LANDING GEAR

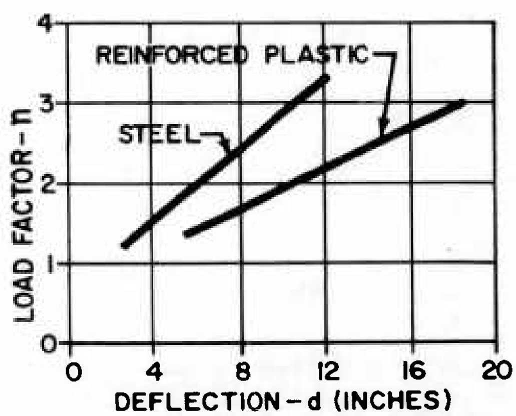


FIGURE 27. LOAD FACTOR VS. DEFLECTION L-19 LANDING GEAR

The estimated weight of the plastic strut is 16 pounds; that of the steel strut is 36 pounds. This is a weight saving of 20 pounds per strut, or 40 pounds per aircraft.

Evaluation

This study has indicated that glass reinforced plastics are feasible material for landing gear struts of the fixed cantilever type. The high strength to weight ratio, good fatigue characteristics and low notch sensitivity make them ideally suited for this application. The steel single leaf spring type of landing gear strut chosen for comparison is simple, lightweight and economical. The strength characteristics of the glass reinforced plastic are such that a strut comparable in strength can be designed for less weight. The curves shown in Figure 27 show that for a given load factor, the deflection of the fiberglass strut is greater than for the metal strut. Figure 28 shows that for a given sinking speed, the developed aircraft load factor is less for the fiberglass strut than for the metal strut. As an example, at a sinking speed of 6 ft./sec., the aircraft load factor developed by this steel strut is 2.75. For the same rate of descent, a similar glass reinforced plastic strut would develop an aircraft load factor of 2.35. Obviously, this will result in less wear and tear on aircraft structure, equipment, and personnel and more comfort for personnel for landing and taxiing operations.

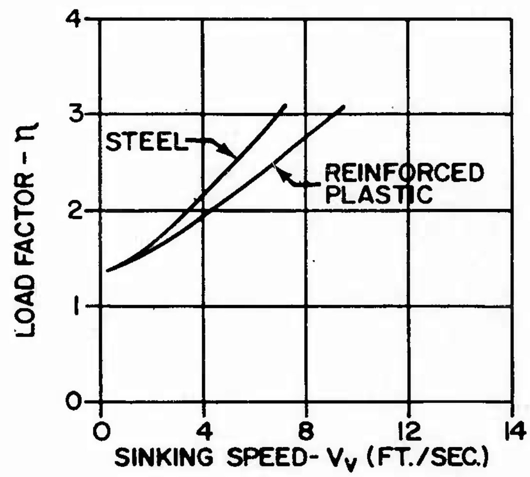


FIGURE 28. LOAD FACTOR VS. SINKING SPEED FOR L-19 AIRCRAFT

The fabrication process for this strut is the same as for the struts for the helicopter previously discussed. It should be molded at moderate to high pressure. Matched metal molds will result in the most satisfactory part if the quantity justifies the increased cost. A female mold with pressure applied with pressure bags will result in satisfactory experimental components and low quantity production.

Filament winding is another feasible method of fabrication. The geometry and cross section would have to be such that the design would be compatible with the process.

The estimated cost for the reinforced plastic strut is $515.00 per unit for a quantity of 10 and $215.00 for a quantity of 100. Per aircraft, cost is twice the figure quoted. The spares cost of the existing metal strut from the Federal Stock Catalog is $45.00 per unit, or $90.00 per

aircraft. This cost comparison is considered to be favorable for reinforced plastics in view of the relative development status of the two types of material.

It is concluded that glass reinforced plastics are feasible materials for cantilevered type landing gear struts for light fixed wing aircraft and other similar applications. The following advantages and disadvantages are summarized.

Advantages

1. Lower aircraft landing load factors for a given sinking speed.

2. Lighter weight strut for a given wheel load.

3. Better shock and vibration damping characteristics, allowing smoother landing and taxiing.

4. Manufacturing methods more adaptable to forming streamlined shapes for increased aerodynamic efficiency.

Disadvantage

Slightly higher initial cost.

Recommendations

It is recommended that development of the reinforced plastic landing gear strut for a specific fixedwing aircraft and/or a helicopter be initiated as expeditiously as practical, to include the following:

1. Make additional analytical studies aimed at sandwich type construction as well as solid laminates and new materials with higher strength to develop a design for a landing gear strut for a particular aircraft.

2. Accumulate data on design and service experience with gear of this type now in use. This has not been possible in this program to date.

3. Accomplish strength and fatigue testing of specimens of beams using construction methods decided upon through analytical studies.

4. Fabricate full-scale components and accomplish strength and fatigue tests.

5. Install reinforced plastic landing gear on aircraft and accomplish drop, flight, and service tests.

POWER TRANSMISSION SHAFT DESIGN STUDY

There are several applications in aircraft where torque tubes or shafts are used to transmit power. These include torque tubes for control systems and drive shafts for rotors and propellers. This study is directed primarily at shafts operating at high speed to determine if reinforced plastics offer any advantages over conventional metal shafts operating at above critical speeds.

The basic, current and foreseeable future design concepts of helicopters utilize a torque tube or shaft to transmit power from the power package to the rotor blade assemblies. These power transmitting shafts are typical of any conventional torque-carrying shaft and have two basic characteristics. First, the applied load is an unsteady load with high frequency oscillations about a mean value; second, they rotate at relatively high speeds. These requirements are of the nature that, due to the inherent quality of energy absorption of reinforced plastics, it is justifiable to consider their use for the fabrication of power transmission shafts.

The general requirements for power transmission shafts in aircraft were used as a basis for this study. These requirements are outlined in References 7, and 47.

Paramount in establishing shaft design is the consideration of fatigue. Torsional vibrations are always present to a greater or lesser degree in all rotating systems, since even the smoothest source generates power in pulses. Since sources of vibrations are always present, the dynamic response of a transmission system to exciting forces must always be investigated if the transmission system is to be considered adequately designed. The dynamic response depends primarily upon the ratio of the exciting frequency to the natural frequency of the shaft and attached masses as well as the severity of the disturbing torque and the damping energy available in the system. In order to avoid undue magnification, the system must be designed so that the natural frequencies avoid as much as possible the major exciting frequencies. This is accomplished by varying the shape and size of the shaft, attached masses and method of support.

High-speed rotation is also of paramount importance and has its primary effect on bearings and supporting structure. All rotating systems have some imbalance. Centrifugal forces produced by this imbalance will eventually induce vibrations which may be transmitted to stationary parts. To keep vibration from exceeding safe levels, operating speeds near the critical-speed range for the system must be avoided. It is desirable that the critical speed be above the maximum operating speed. However, in many cases it is impractical to design for such a condition. Under this condition, the critical speed is established well below the minimum operating speed so that the critical speed is passed through in arriving at the operating speed. The following discussion of the critical speed of shafts is included as a basis for evaluation of the effect of material properties on this phenomenon.

In order to simplify the discussion and analysis, the configuration considered is one of a slender shaft with a circular disc at the center of the length. Although this assumed configuration is not sufficient for the analysis of a particular shaft with a distributed mass, it provides a simple means of studying the phenomenon under consideration.

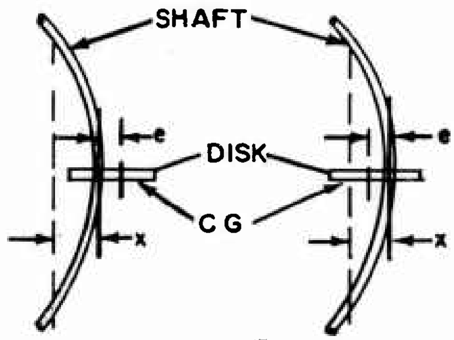


FIGURE 29. CRITICAL SHAFT SPEED SCHEMATIC

At shaft rotation speeds up to the critical speed, the shaft deflects as indicated in Figure 29. The equilibrium equation is:

$$\frac{W}{g}(x + e)\,\omega^2 = kx$$

where

W = weight of disc, lb.

g = acceleration of gravity, in./sec.^{2}

x = deflection, in.

e = eccentricity, in.

ω = speed of rotation of shaft, rad./sec.

k = spring constant, lb./in.

Solving for the displacement we have

$$x = \frac{e}{\frac{k}{\omega^2}\,\frac{g}{W} - 1}$$

Denoting the natural frequency of the system by

$$p^2 = \frac{kg}{W}$$

then

$$x = \frac{e}{\frac{p^2}{\omega^2} - 1}$$

It is seen that as the rotational frequency, ω, approaches the natural frequency, p, the displacement, x, increases rapidly. The critical speed occurs when the two frequencies become equal.

At rotational speeds above the critical speed, experiment shows that the center of gravity is situated between the axis of rotation and the deflected axis of the shaft as indicated in Figure 29, so that the equilibrium equations becomes

$$\frac{W}{g}(x - e)\omega^2 = kx$$

or

$$x = \frac{e}{1 - \frac{kg}{\omega^2 W}} = \frac{e}{1 - \frac{p^2}{\omega^2}}$$

With increasing speed, ω, the deflection, x, becomes smaller.

From these formulations, it can be seen that a smaller eccentricity, e, will result in a slower build up of displacement, x, below the critical speed and a more rapid decline of x above the critical speed; but it would have no effect for prolonged operation at the critical speed. The natural frequency defines the speed at which the critical speed occurs.

For rotational speeds at or below the critical speed, damping has no effect except to damp out disturbances to the stable condition formulated above. However, at speeds above the critical speed, damping due to hysteresis can, following a disturbance, maintain a condition where the plane of the deflected shaft is rotating at critical speed while the shaft itself is rotating at a greater speed. This condition, known as whirling, is maintained as a result of the fact that the axis of zero stress does not coincide with the axis of zero strain within a given cross section of the deflected shaft. This condition is, in turn, a result of the fact that due to hysteresis the stress-strain relationship in going from a compressive stress to a tension stress is not the same as the relationship in the reversed condition. Under these conditions, an unlimited increase in deflection of the shaft can occur at shaft rotational speeds above the critical speed. For a further discussion of this phenomenon, refer to Timoshenko's <u>Vibration Problems in Engineering.</u>

As a result of the foregoing analysis, the following conclusions can be made relative to the use of reinforced plastics in power transmission shafts.

1. The increased damping of reinforced plastics as compared to aluminum will result in the occurrence of whirling over a broader range of shaft speed.

2. The lower stiffness to weight ratio of plastics results in lower natural frequencies and, therefore, lower critical speeds. However, the amplitude and resulting stresses are a function of the ratio of the critical speed to shaft speed and not critical speed alone. Therefore, assuming a proper design with the operating speed sufficiently remote from the critical speed, the actual value of the critical speed does not significantly affect shaft life.

Experience has shown that the primary consideration in the design of a power transmission shaft is fatigue life. Therefore, the operating stresses must be kept relatively low. S/N curves for metals are readily available for use in predicting the life of metal drive shafts. Corresponding data for reinforced plastic drive shafts are not available. Therefore, the fatigue life of fiberglass reinforced plastics was approximated by using relative values of axial fatigue data for reinforced plastics and aluminum. This method is not considered to be adequate for the purpose. It is the best approach that could be taken with available information. Complete fatigue tests should be made on any intended application for transmission shafts.

To evaluate the use of reinforced plastics in power transmission shafts, two typical applications were investigated and compared with existing metal shafts for these applications. These examples are the HU-1 tail rotor drive shaft and the shaft between the main transmission and the rotor transmission of the H-21 helicopter. These two examples were chosen because they represent a wide variation in transmitted power and size of shaft required.

Since fatigue life is of primary concern, a preliminary design of a fiberglass reinforced plastic shaft was made for each example that would have the same estimated life as the existing metal shaft. Stresses for the metal shaft were determined for a load corresponding to a condition of maximum operating power and the associated rotational speed. These stresses, in conjunction with appropriate S/N curves, were used to determine the estimated service life. Using this analogy, the estimated service life was determined to be approximately 10 million cycles for both shafts.

A summary of the results of the two applications is shown in Table 6.

The reinforced plastic shaft can be fabricated by either filament winding or with a cloth lay-up. Filament winding is preferred because of higher strength and lower cost. The cloth lay-up should be cured at moderately high pressure to insure maximum strength characteristics for this method. Epoxy resin should be used for maximum strength.

The comparison in Table 6 shows that the fiberglass shafts are slightly heavier than aluminum shafts. This comparison is for only the typical shaft section. End attachments to fittings and bearings will be more complex in the fiberglass shaft because of the low bearing strength. An increase in thickness or a metal insert at the ends will be required to increase the strength for mechanical attachments.

EVALUATION

The reinforced plastic shafts have about the same fatigue strength to weight ratio as the aluminum alloy shaft. This assumption is based on relative fatigue values for axially loaded specimens since torsion fatigue values are unavailable. Prior to utilization of plastics for transmission shafts, fatigue tests of this configuration of loading and structure should be performed to substantiate this condition.

TABLE 6

POWER TRANSMISSION SHAFT SUMMARY

	HU-1	H-21
Design Data		
Shaft Speed - r.p.m.	3780	2500
Shaft Torque - in.-lb.	1250	16,150
Power - SHP	75	1280
Aluminum Shaft		
Material	2024S-T4	2024S-T4
Outside Diameter - in.	3.02	4.10
Wall Thickness - in.	.050	.15
Length - in.	281	240
Working Stress - p.s.i.	2880	4560
Weight per Linear Inch - lb.	.0231	.196
Reinforced Plastic Shaft		
Material	Filament Wound Epoxy Resin	
Outside Diameter - in.	3.02	4.10
Wall Thickness - in.	.080	.26
Length - in.	281	240
Working Stress - p.s.i.	2300	2800
Weight per Linear Inch - lb.	.0235	.242

The basic plastic tube has a greater degree of unbalance as fabricated than the aluminum tube and provides greater difficulty in balancing.

The attachment of end fittings to accommodate quick disconnects, bearings and flexible couplings is also somewhat more difficult with plastics because of low bearing allowables for attachments.

Although reinforced plastics have a greater degree of internal damping than metals, this property has negligible effect in the fatigue life under the particular conditions of forced vibrations at frequencies appreciably different from natural frequencies. This increased damping can cause a detrimental condition known as whirling at supercritical speeds.

The power transmission shaft designed in plastics is more flexible than one of aluminum, resulting in lower natural frequencies. This decrease in natural frequency is not considered to be critical except in those cases where it may be possible to maintain a critical speed above operating speeds when designed in aluminum.

Since the fabrication of metal transmission shafts is relatively simple, the reinforced plastic shaft is appreciably more costly using present fabrication techniques.

In summarizing this evaluation, the use of reinforced plastics for power transmission shafts has the following advantages and disadvantages:

Advantages

1. Greater resistance to environmental conditions
2. Radar transparent

Disadvantages

1. More susceptible to whirling
2. Greater unbalance requiring correction
3. More development required
4. Increased cost
5. Complicated detail design

The disadvantages of using reinforced plastics for power transmission shafts do not prohibit their use in this application. Their only real advantages are radar transparency and greater resistance to environmental conditions. It is therefore concluded that reinforced plastics are not feasible materials for use in power transmission shafts unless their special characteristics make their use mandatory.

TRANSMISSION HOUSING DESIGN STUDY

Transmission housings are found on virtually every type of Army aircraft. They enclose a system of gear trains; provide a means of lubricating precision bearings or gears; provide a suitable structure for the transfer of such loads as thrust, lift, and torque to the airframe; and provide for the dissipation of heat generated during transmission operation. The size and complexity of the transmission and its housing are largely determined by the amount of power to be transmitted, the degree of speed reduction, and the number of output drive shafts. All these factors pose problems in the design of a suitable transmission assembly for helicopters, since they usually involve large amounts of power and large speed reduction and use a common power source for a number of different functions.

Because of the large amount of gearing involved in helicopter transmissions, it is believed that the transmission assembly is the source of considerable vibration and noise. This can cause fatigue stresses in critical structures and uncomfortable noise levels inside the helicopter crew area. These adverse effects could be reduced by using a vibration isolating material such as reinforced plastic to fabricate the transmission housings.

In general, present transmission designs are of cast aluminum or magnesium materials, which offer very little vibration damping. These materials, however, have good mechanical-physical properties. It is the purpose of this study to investigate the feasibility of using reinforced plastic materials for transmission housings to replace presently used metal alloys.

The investigation includes an analysis of general problem areas in transmission housings and the mechanical-physical properties of reinforced plastic materials and light metal alloys. Problem areas are listed below, and succeeding paragraphs discuss the comparative mechanical-physical properties of the various materials. Concluding paragraphs present the relative merits of using reinforced plastic materials for transmission housings.

Design requirements must be satisfied under conditions of extreme mechanical vibrations and those environmental conditions stipulated in MIL-T-5955B. This specification requires operation over an ambient temperature range of -65° to 160°F and in other environmental conditions common to world-wide military operations.

The selection of housing material to meet the transmission requirements is based on the following desirable characteristics:

1. Good dimensional stability under load, high modulus of elasticity, low creep strain and low coefficient of thermal expansion.

2. High fatigue strength to weight ratio and correspondingly high values of ultimate, yield, and creep strength.

3. Economical fabrication and material costs - ease of forming into irregular shapes and providing for multiple driven shafts of accessory power shafts.

4. Good damping characteristics.

5. High thermal conductivity for dissipation of heat.

6. Good corrosion resistance and chemical compatibility with a wide range of lubricants.

A comparative analysis was conducted of these characteristics for materials currently used in transmission housings and for a number of reinforced plastics materials which may be used. Values for each of the pertinent characteristics are tabulated in Table 7. The basis for each of the characteristics is discussed in the following paragraphs.

MECHANICAL PROPERTIES OF MATERIALS

The tensile properties, i.e., ultimate strength, yield strength, fatigue strength, and tensile elastic modulus of the various potential transmission housing materials, were considered for comparative purposes because they appear to be representative of compressive, flexural, and shear properties. Wherever test data were available, the ultimate strength and yield strength were obtained from the specimens used in obtaining the fatigue or creep data.

The yield strength for the reinforced plastic materials is not well defined because the material does not yield as many metals do. There are sometimes two or more distinct slopes to the stress-strain curves. In these cases, the proportional limit for the initial slope is used as the yield strength along with the corresponding elastic modulus. All properties for the reinforced plastic materials are for the direction parallel to the principal reinforcing fibers.

The fatigue strength data shown in Table 7 were obtained from direct tests which most nearly represent the stress condition in this application. Since most available reinforced plastic fatigue data are of the direct stress type and most available fatigue data of light metal alloys are of the rotating beam type, direct comparison of fatigue properties of the two was difficult because the data were not quantitatively comparable. Available comparable fatigue data are shown in Figures 30 through 34. From the limited fatigue data available, however, it appears that the fatigue strength of the reinforced plastics is superior to aluminum sand castings and inferior to or possibly equivalent to magnesium castings.

Characteristics of reinforced plastics vary widely with different resins and reinforcing weaves. The effects of different resins can be seen by comparing the characteristics of 181 epoxide and 181 heat resistant polyester. Epoxide resin improves fatigue strength by approximately 75 percent.

Based on this comparison and noting the comparative fatigue strength of the magnesium castings and 181 polyester laminate in Figure 34, it is believed that the 181 epoxide fatigue characteristics would be on a par with the magnesium castings (particularly the AZ92A-T6 and the AZ63A-T6 alloys). The effects of the reinforcement on the fatigue characteristics can be seen in Figures 30 through 34.

In studying the direct stress fatigue characteristics of the various reinforced plastics, it was found that the characteristics for the wet and dry conditions for certain laminates varied considerably. For most of the laminates, the fatigue strength for the wet condition was considerably less than that for the dry condition, but the wet and dry fatigue strength for the 181 epoxide laminate was approximately the same. It appears that the type of resin is a major contributing factor in the wet and dry fatigue strength.

The fatigue strength to density ratio for comparing various materials is tabulated in column 7 of Table 7. The higher this value, the more desirable the particular material is from a weight standpoint. When considering the wet condition characteristics of the various reinforced plastics (the wet and dry strengths of metals are identical), the 181 epoxide laminate was the lightest and the asbestos mat-phenolic was the next lightest. Both were comparatively lighter than the 2024-T4 aluminum.

In the dry condition, it was noted that the unwoven crossply laminate had the greatest strength to density ratio, followed by the phenolic asbestos mat and the 181 epoxide laminates in that order. No dry condition data were available on the unwoven parallel laminate, but it is believed that it would have an even higher strength to density ratio.

Because of the lack of fatigue data, only the 181 heat resistant polyester reinforced plastic could be directly compared with the magnesium castings. Since the density of the two materials is approximately the same and the fatigue strenth of the magnesium castings is considerably greater, the latter is somewhat lighter. However, since 181 epoxide was superior to the 181 polyester in the zero mean stress level tests, the 181 epoxide may be on a par with the magnesium castings from a weight standpoint.

The relative stiffness of the various materials considered was compared using a "specific stiffness factor", or the ratio of the tensile elasticity modulus to the density, Table 7, Column 8. This ratio is a measure

of dimensional stability and, consequently, high values are desirable. The specific stiffness of aluminum and magnesium was found to be much higher than any of the reinforced plastic materials; but of the reinforced plastics, the unwoven parallel and the asbestos laminates had superior stiffness characteristics, generally 50 percent greater than the other laminates. There does not appear to be any appreciable difference between the reinforced plastic specific stiffness values for the wet and dry conditions.

The tensile rupture strength of the various materials was compared from the data in column 9 of Table 7. The 1,000-hour tensile rupture strength at 300°F was found to be considerably greater than the direct stress fatigue strength for most stress. The 1,000-hour tensile rupture strength for the reinforced plastic laminates was approximately the same, ranging from 24,000 to 40,000 p.s.i. For some laminates, namely 181 silicone, 181 CTL-91LD (phenolic), phenolic asbestos mat and unwoven isotropic laminates, increasing the temperature to 300°F had very little effect on the 1,000-hour stress-rupture strength. For these laminates, the 1,000-hour rupture strength ranged from 90 percent to 100 percent of the room temperature values. Other laminates (181 heat-resistant polyester, unwoven parallel epoxy, and unwoven crossply-epoxy) ranged from 65 percent to 80 percent. No comparable data were available for the remaining laminates or the light metal alloys.

Creep strain data, measured at a stress level slightly less than the 1,000-hour tensile rupture, column 10 of Table 7, indicate that the reinforced plastic laminates and aluminum had very little creep after 1,000 hours under load and at temperatures up to 300°F. Most of the reinforced plastics had a lower creep strain ratio than the aluminum, even though the reinforced plastic was subjected to over twice the initial strain of the aluminum. The strain ratio for magnesium was 7 to 9 times that of the reinforced plastics. Thus, it appears that the reinforced plastic laminates and aluminum have good creep characteristics as compared to magnesium.

TABLE 7
COMPARATIVE MECHANICAL AND PHYSICAL PROPERTIES
REINFORCED PLASTICS AND LIGHT WEIGHT METAL ALLOYS

Material	Tensile Ultimate Strength (A) 1000 p.s.i.		Tensile Yield Strength or Proportional Limit (A) 1000 p.s.i.		Direct Stress (F) Fatigue Strength (A) 50×10^6 Cycles 1000 p.s.i.		Tensile Elasticity Modulus (A) 10^6 p.s.i.		Density ($lb/in.^3$)
	Dry	Wet	Dry	Wet	Dry	Wet	Dry	Wet	
REINFORCED PLASTICS									
181 Polyester	46.0	42.3	9.45	-	9.5	8.3	3.23	-	.0675
181 Heat Resist Polyester	46.0	-	18.1	-	12.6(B)	-	2.65	-	.0685
181 Epoxy	48.2	43.4	13.8	17.1	14.2	11.9	3.58	3.34	.0665
181 Epoxide	41.2	44.6	16.3	-	14.0	15.6	3.04	-	.0650
181 Heat Resist Epoxide	43.1	-	13.8	-	7.8	-	2.81	-	.0675
Unwoven Parallel	141.9	131.6	70.2	75.5	-	8.5	5.05	4.69	.0651
Unwoven Crossply	63.9	59.4	17.6	15.2	19.7	8.5	3.36	3.45	.0695
Unwoven Isotropic	-	51.1	40.7	36.7	-	7.2	2.56	2.54	.0685
181 Phenolic	45.1	-	12.8	-	10.2	-	3.46	-	.0693
181 Silicone	35.3	-	21.0	-	6.2	-	2.64	-	.0668
Asbestos Mat Phenolic-Parallel	48.6	42.8	26.8	-	16.7	15.0	5.10	-	.0679
Asbestos Mat Phenolic-Cross	41.5	38.4	-	-	12.7	11.0	-	-	.0675
Fiberglass Mat-Polyester	13.9	-	5.17	-	2.8	-	1.10	-	.0520
181 CTL-91LD Phenolic	45.9	-	12.5	-	-	-	4.02	-	.065-.076(C)

TABLE 7 (CONT'D)
COMPARATIVE MECHANICAL AND PHYSICAL PROPERTIES
REINFORCED PLASTICS AND LIGHT WEIGHT METAL ALLOYS

Material	Tensile Ultimate Strength (A)		Tensile Yield Strength or Proportional Limit (A)		Direct Stress (F) Fatigue Strength (A) 50×10^6 Cycles		Tensile Elasticity Modulus (A)		Density
	1000 p.s.i.		1000 p.s.i.		1000 p.s.i.		10^6 p.s.i.		
	Dry	Wet	Dry	Wet	Dry	Wet	Dry	Wet	(lb./in.^{3})
MAGNESIUM									
AZ63A-T6 Casting	34.0	-	16.0	-	23.2(B)	-	6.50	-	.0656
AZ92A-T6 Casting	34.0	-	18.0	-	27.0(B)	-	6.50	-	.0659
AZ80A-T5 Forging	42.0	-	28.0	-	-	-	6.50	-	.0649
ALUMINUM									
195-T6 Casting	32.0	-	20.0	-	6.0	-	9.9	-	.102
355-T6 Casting	41.0	-	31.0	-	6.0	-	10.1	-	.098
356.T6 Casting	38.0	-	28.0	-	6.0	-	10.4	-	.097
2024-T4 Alloy Plate	65.0	-	48.0	-	18.5	-	10.5	-	.100

TABLE 7 (CONT'D)
COMPARATIVE MECHANICAL AND PHYSICAL PROPERTIES
REINFORCED PLASTICS AND LIGHT WEIGHT METAL ALLOYS

Material	Fatigue Strength To Density Ratio (A)		Specific Stiffness (Modulus To Density Ratio) (A)		Tensile Rupture Strength @ 1000 Hr. 300°F (A)	Creep Strain @ 1000 Hr. 300°F (A)		Coefficient of Thermal Expansion Warp Direction
	10^5 In.		10^8 In.			Total	Ratio	
	Dry	Wet	Dry	Wet	1000 p.s.i.	%		10^{-6} In./In. °F
REINFORCED PLASTICS								
181 Polyester	1.40	1.23	.48	-	25.8 (Dry)	1.17 Dry	1.14 Dry	7.8 (-100 to +100°F)
181 Heat Resist Polyester	1.84(B)	-	.39	-	23.5(D)	1.44 (D)	1.45 (D)	5.5 (-100 to +200°F)
181 Epoxy	2.13	1.79	.54	.50	32.4 (Dry)	1.26 Dry	1.15 Dry	5.5 (-100 to +200°F)
181 Epoxide	2.15	2.40	.47	-	-	-	-	6.5 (-100 to +100°F)
181 Heat Resist Epoxide	1.15	-	.42	-	-	-	-	-
Unwoven Parallel	-	1.30	.78	.72	40.0(250°F)	-	-	4.8 (-30 to +200°F)
Unwoven Crossply	2.83	1.22	.48	.50	35.0(250°F)	-	-	7.1 "
Unwoven Isotropic	-	1.04	.37	.37	36.0(250°F)	1.80 (250°F)	1.12 (250°F)	8.4 "
181 Phenolic	1.47	-	.50	-	-	-	-	6.0 (-100° to +200°F)
181 Silicone	.93	-	.40	-	24.0	1.06	1.08	4.3 (-100 to +100°F)
Asbestos Mat Phenolic-Parallel	2.46	2.20	.75	-	38.0	1.055	1.52	2.6 (-100 to +100°F)

TABLE 7 (CONT'D)
COMPARATIVE MECHANICAL AND PHYSICAL PROPERTIES
REINFORCED PLASTICS AND LIGHT WEIGHT METAL ALLOYS

Material	Fatigue Strength To Density Ratio (A)		Specific Stiffness (Modulus To Density Ratio) (A)		Tensile Rupture Strength @ 1000 Hr. 300°F (A)	Creep Strain @ 1000 Hr. 300°F (A)		Coefficient of Thermal Expansion Warp Direction
	10^5 In.		10^8 In.			Total	Ratio	
	Dry	Wet	Dry	Wet	1000 p.s.i.	%		10^{-6} In./In.-°F
Asbestos Mat	1.88	1.63	-	-	-	-	-	-
Phenolic-Cross Fiberglass Mat-Polyester	.54	-	.21	-	11.6 (Dry)	0.76 Dry	1.68 Dry	-
181 CTL-91LD	-	-	.53-.62	-	32.0	1.17	1.19	6.0 (-100 to +200°F)
MAGNESIUM								
AZ63A-T6 Casting	3.54(B)	-	.99	-	8.0	1.0 0.5	10.0 7.7	14.0(65-212°F
AZ92A-T6 Casting	3.95(B)	-	.99	-	-	-	-	14.0 "
AZ80A-T5 Forging	-	-	1.00	-	9.9	0.5	8.3	14.0 "
ALUMINUM								
195-T6 Casting	.59	-	.97	-	-	-	-	12.7(68.212°F
355-T6 Casting	.61	-	1.03	-	36.0	.5	1.40	12.4 "
356-T6 Casting	.62	-	1.07	-	24.0	.5	2.18	11.9 "
2024-T4 Alloy Plate	1.85	-	1.05	-	-	-	-	12.6 "

TABLE 7 (CONT'D)
COMPARATIVE MECHANICAL AND PHYSICAL PROPERTIES
REINFORCED PLASTICS AND LIGHT WEIGHT METAL ALLOYS

Material	Specific Heat	Thermal Conductivity	Data Reference
	BTU/Lb.-°F	BTU-In./Hr.-Ft.^{2}-°F	
REINFORCED PLASTICS			
181 Polyester	.25 (100°F)	.9 (100°F)	16, 38
181 Heat Resist Polyester	.27 (100°F)	1.1 (100°F)	16, 28, 38
181 Epoxy	0.3	1.2 (100°F)	17, 38
181 Epoxide	-	1.12 (100°F)	16, 38, 58
181 Heat Resist Epoxide	-	1.2 (100°F)	16, 58
Unwoven Parrallel	.21	-	17, 54
Unwoven Crossply	.21	-	17, 54
Unwoven Isotropic	.21	2.37 (125°F)	17, 54
181 Phenolic	.23 (100°F)	.35 (100°F)	16, 38
181 Silicone	.246	0.62 (100°F)	16, 18, 57
Asbestos Mat Phenolic-Parallel	.31	1.30 (100°F)	17, 20, 31
Asbestos Mat Phenolic-Cross	-	-	17
Fiberglass Mat-Polyester	-	-	16
181 CTL-91LD Phenolic	.273	1.10 (100°F)	19, 31

TABLE 7 (CONT'D)
COMPARATIVE MECHANICAL AND PHYSICAL PROPERTIES
REINFORCED PLASTICS AND LIGHT WEIGHT METAL ALLOYS

Material	Specific Heat	Thermal Conductivity	Data Reference
	BTU/Lb.-°F	BTU-In. / Hr.-Ft.² - °F	
MAGNESIUM			
AZ63A-T6 Casting	.25 (78°F)	529	37
AZ92A-T6 Casting	.25 "	468	37
AZ80A-T5 Forging	.25 "	529	37
ALUMINUM			
195-T6 Casting	.23 (212°F)	960 (77°F)	37, 72
355-T6 Casting	.23 "	985 "	24, 37, 72
356-T6 Casting	.23 "	1100 "	24, 37, 72
2024-T4 Alloy Plate	.23 "	840 "	2, 37

Notes:
A. Dry Condition -73°F and 50% Humidity; Wet Condition - 100°F and 100% Humidity.
B. Based on direct stress fatigue tests with a minimum to maximum stress ratio of 1/4.
C. Test specimens for strength and creep data had an average density of .065 lb/in^3; test specimens for thermal data had an average density of .075 lb/in^3.
D. Based on data from tests of Vibrin 135 TAC Polyester Laminates, Reference 28.

TABLE 8
PERTINENT MATERIAL DATA FOR
REINFORCED PLASTICS FATIGUE AND CREEP TESTS

Designation	Reinforcement Type	Reinforcement Finish	Resin & Content	Lay-up and Laminating Pressure	Test (4) Condition	Ref.
181-Polyester	181	Volan A	34.3% Paraplex P-43 (Forest Products Laboratory	Parallel 14 p.s.i	73/50 & 100/100	16
181-Epoxide	181	Volan A	37.7% Epon 828-14% CL (Shell Development Co.)	Parallel 50 p.s.i.	73/50 & 100/100	16
Mat-Polyester	Mat-1½ oz. Glass	-	58.3% Paraplex P-43	Parallel 14 p.s.i.	73/50	16
181-Epoxy	181	Volan A	26.8% Scotchply 1002 Resin	Parallel 30 p.s.i.	73/50 & 100/100	17
Unwoven Parallel	Scotchply Type 1002		31.5% Avg.	Parallel 30 p.s.i.	100/100	17
Unwoven Crossply	Scotchply Type 1002		30.9% Avg.	Cross-laminate 30 p.s.i.	73/50 & 100/100	17
Asbestos Mat-Phenolic	Raybestos Manhattan Asbestos Mat (4 & 8 oz. Pyrotex)		29.1% Avg. R/M Phenolic Resin Style 984-RPD	Parallel (8oz.) and Cross Lam. (4 oz.) 400 p.s.i.	73/50 & 100/100	17
181 Polyester-Heat Resistant	181	Volan A	35.3% PDL 7-669 (Forest Products Laboratory)	Parallel 14 p.s.i.	73/50	16
181 Epoxide-Heat Resistant	181	Volan A	35% Epon X12100-4%E (Shell Development Co.)	Parallel 200 p.s.i.	73/50	16

TABLE 8 (CONT'D)
PERTINENT MATERIAL DATA FOR
REINFORCED PLASTICS FATIGUE AND CREEP TESTS

Designation	Reinforcement		Resin & Content	Lay-up and Laminating Pressure	Test (4) Condition	Ref.
	Type	Finish				
181 Phenolic	181	Volan A	28% BV 17085 (Swedlow Plastics Co.)	Parallel	73/50	16
181 Silicone	181	Heat Cleaned	31% DC 2106 (Dow Corning Co.)	Parallel	73/50	16
181 CTL-91LD Phenolic	181	A 1100	37.3% CTL-91LD (Cincinnati Testing & Research Labs. Inc.)	Parallel 200 p.s.i.	300°F. (Creep)	19
181 Vibrin 135 Polyester (TAC)	181	United Merchants Corp. Type 301	37% Vibrin 135 (CN-7252) (Naugatuck Chemical Div., U.S. Rubber Co.)	Parallel 15-20 p.s.i.	300°F. (Creep)	28

Notes: (1) All tests conducted on a direct-stress fatigue machine. (Fatigue Tests)
(2) All specimens were unnotched. (Fatigue Tests)
(3) All specimens listed tested at 0 mean stress, 900 cycles/min., and zero angle of loading relative to surface ply. (Fatigue Tests)
(4) Temperature °F and relative humidity %

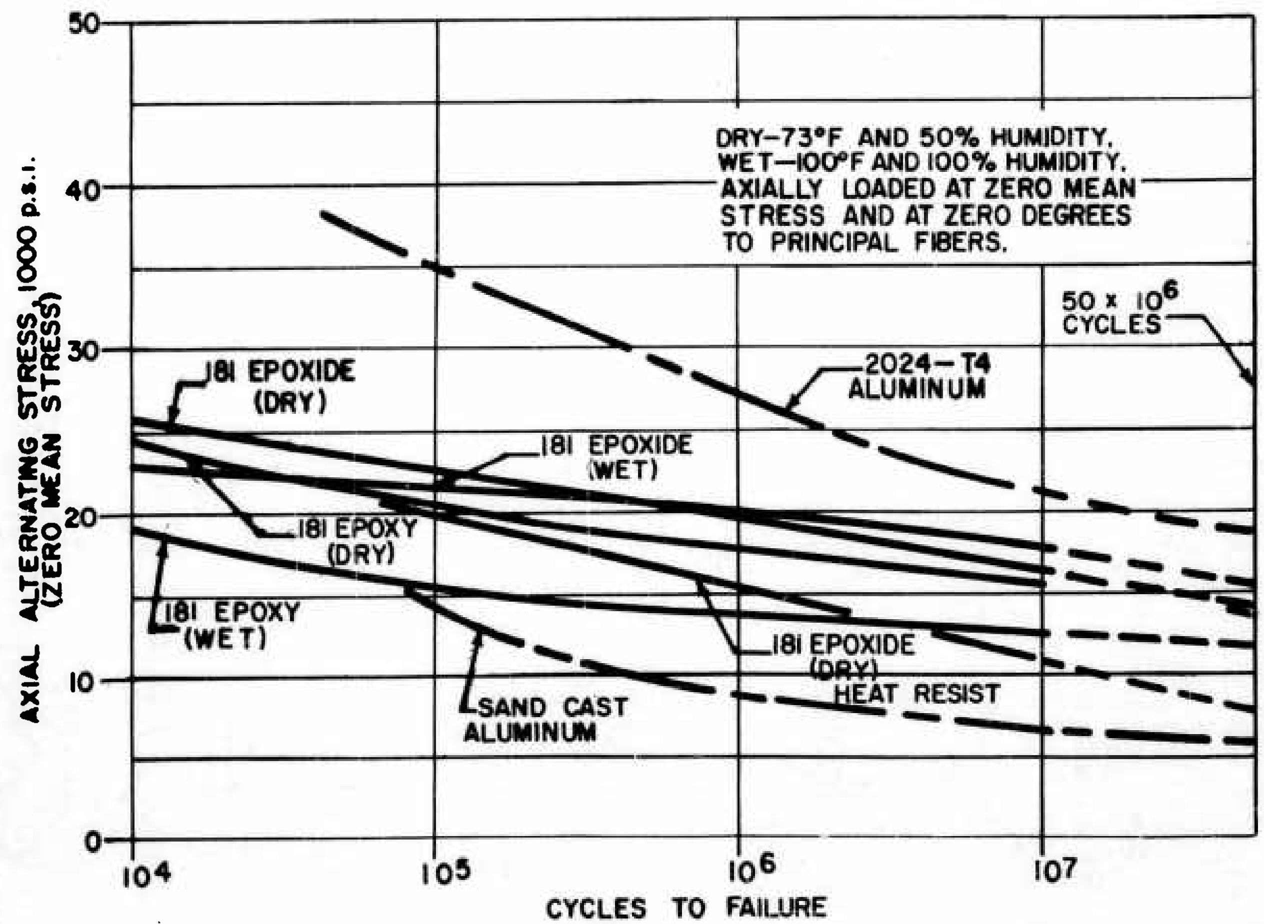


FIGURE 30. DIRECT STRESS FATIGUE CHARACTERISTICS -- ALUMINUM ALLOY AND WOVEN TYPE 181 GLASS FABRIC PLASTIC LAMINATES

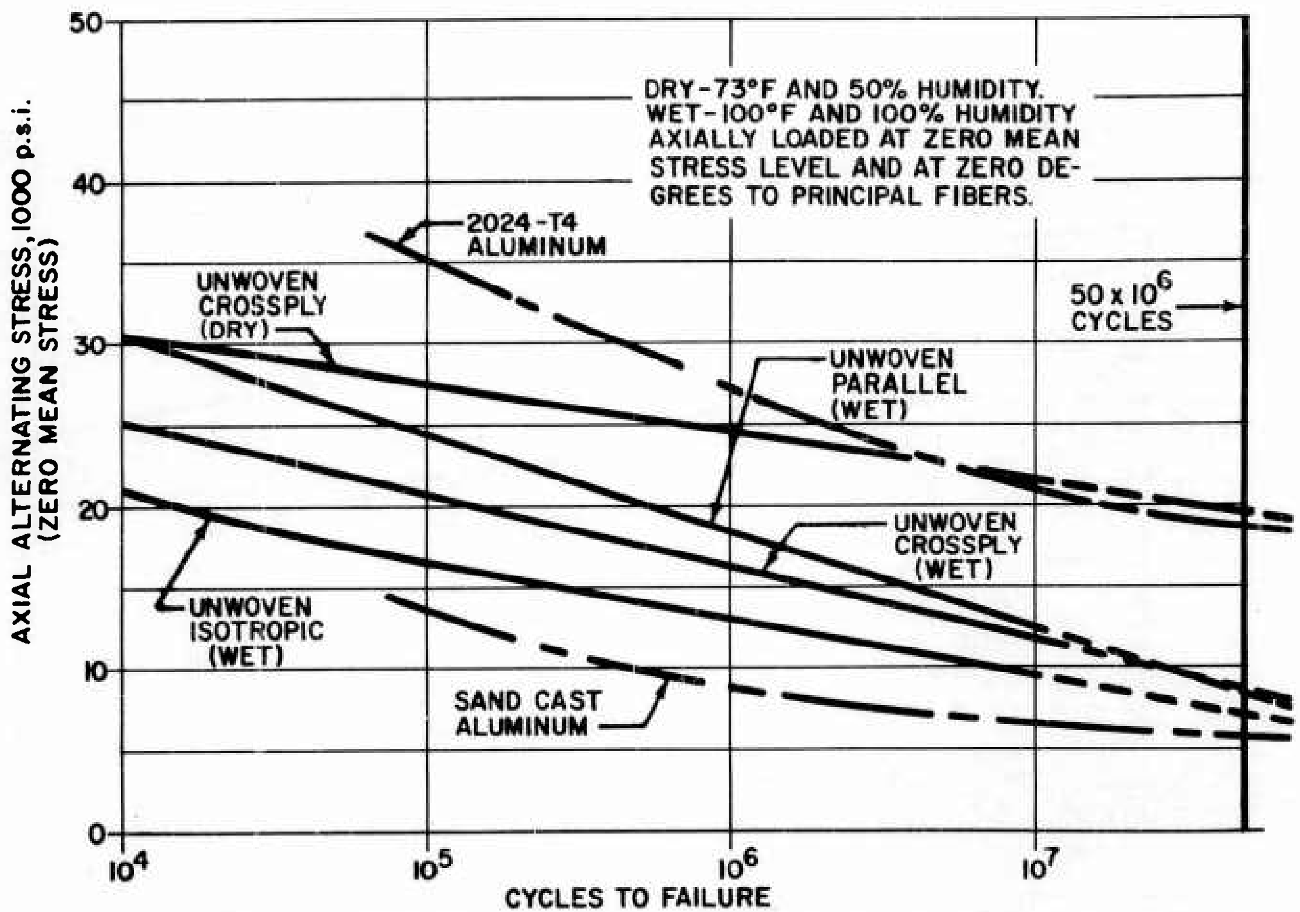


FIGURE 31. DIRECT STRESS FATIGUE CHARACTERISTICS – ALUMINUM ALLOY AND UNWOVEN GLASS FIBER PLASTIC LAMINATES

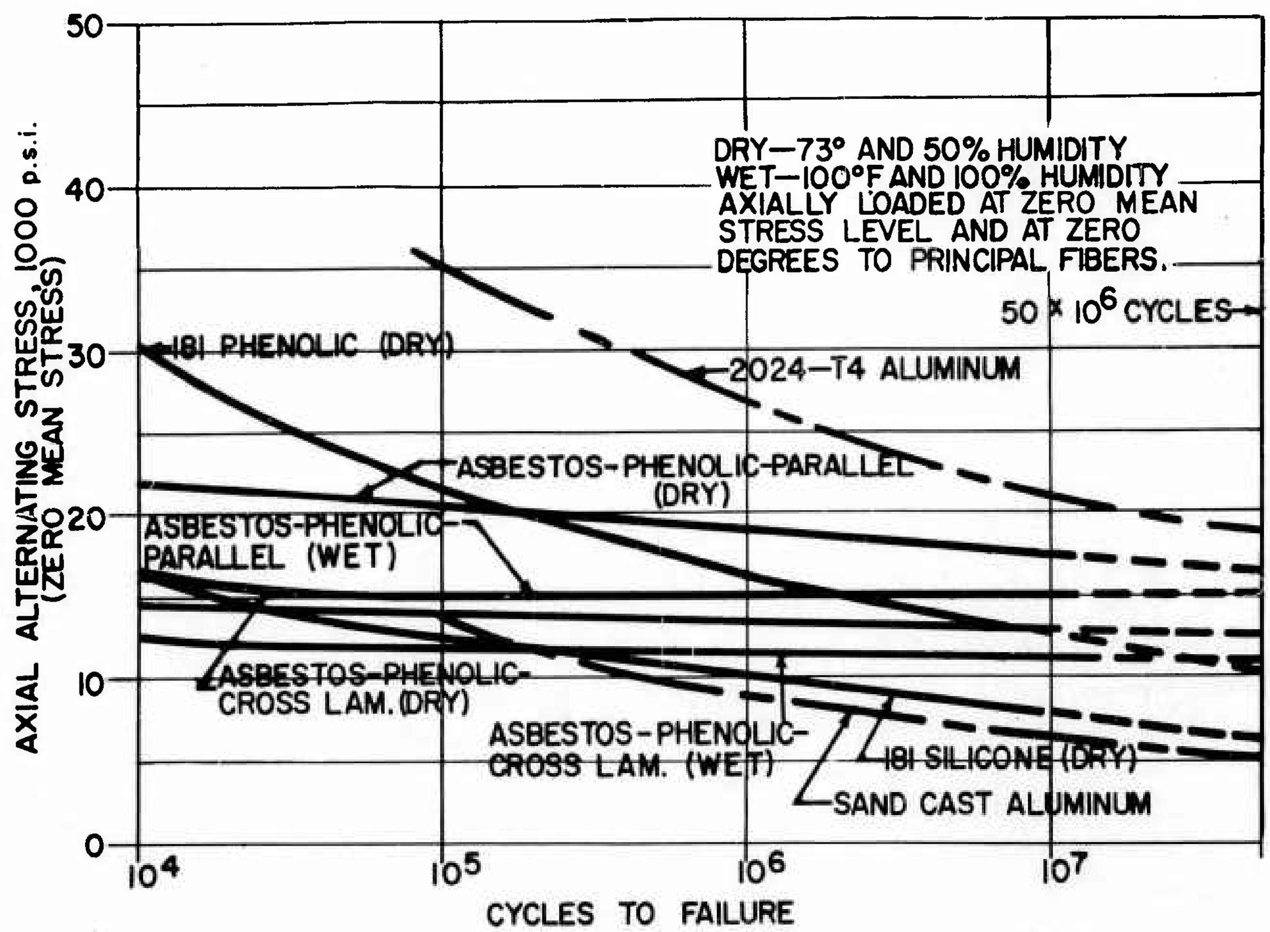


FIGURE 32. DIRECT STRESS FATIGUE CHARACTERISTICS – ALUMINUM ALLOY VS. ASBESTOS AND GLASS FIBER PLASTIC LAMINATES

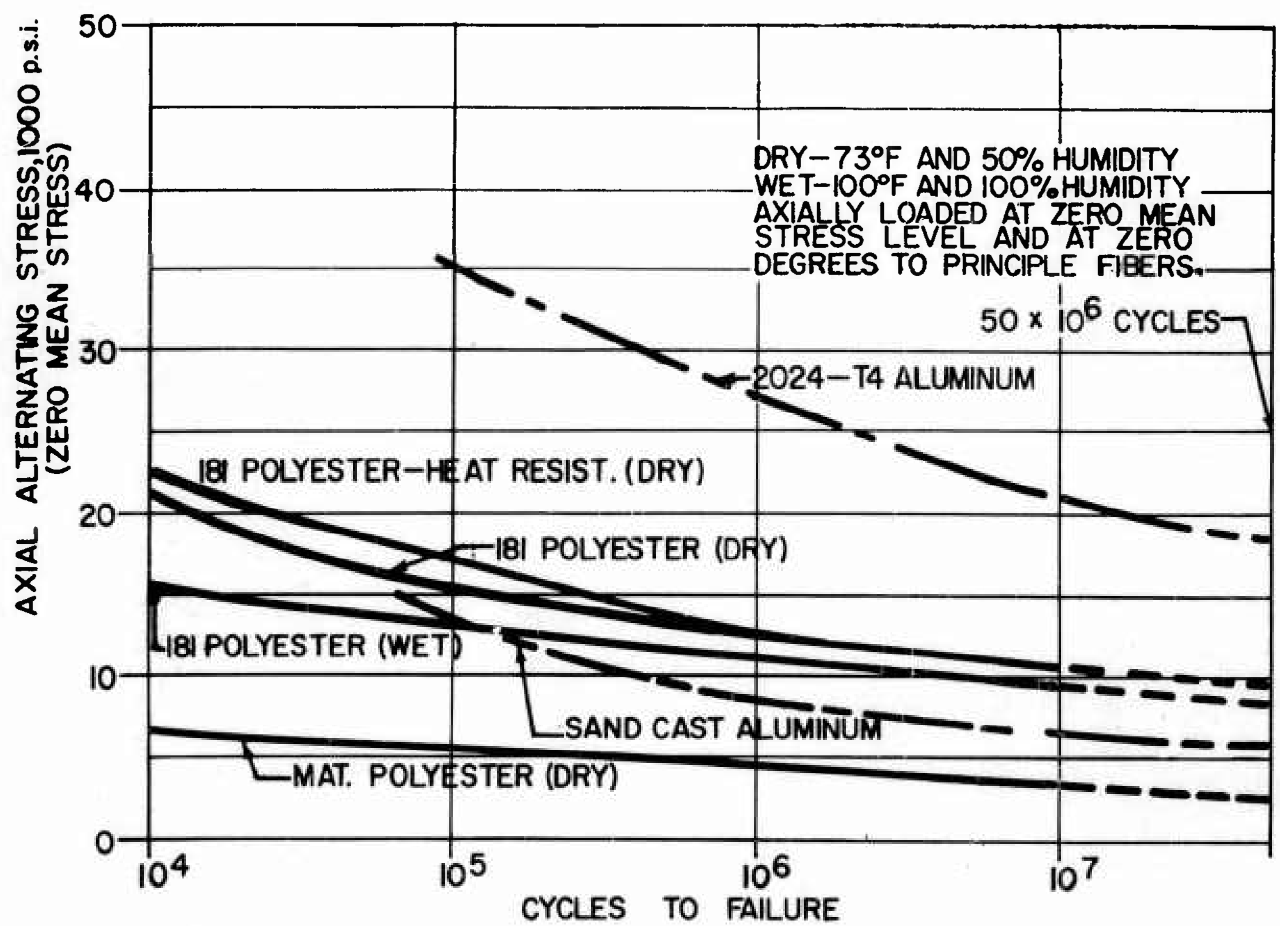


FIGURE 33. DIRECT STRESS FATIGUE CHARACTERISTICS – ALUMINUM ALLOY VS. GLASS FIBER MAT AND WOVEN FABRIC PLASTIC LAMINATES

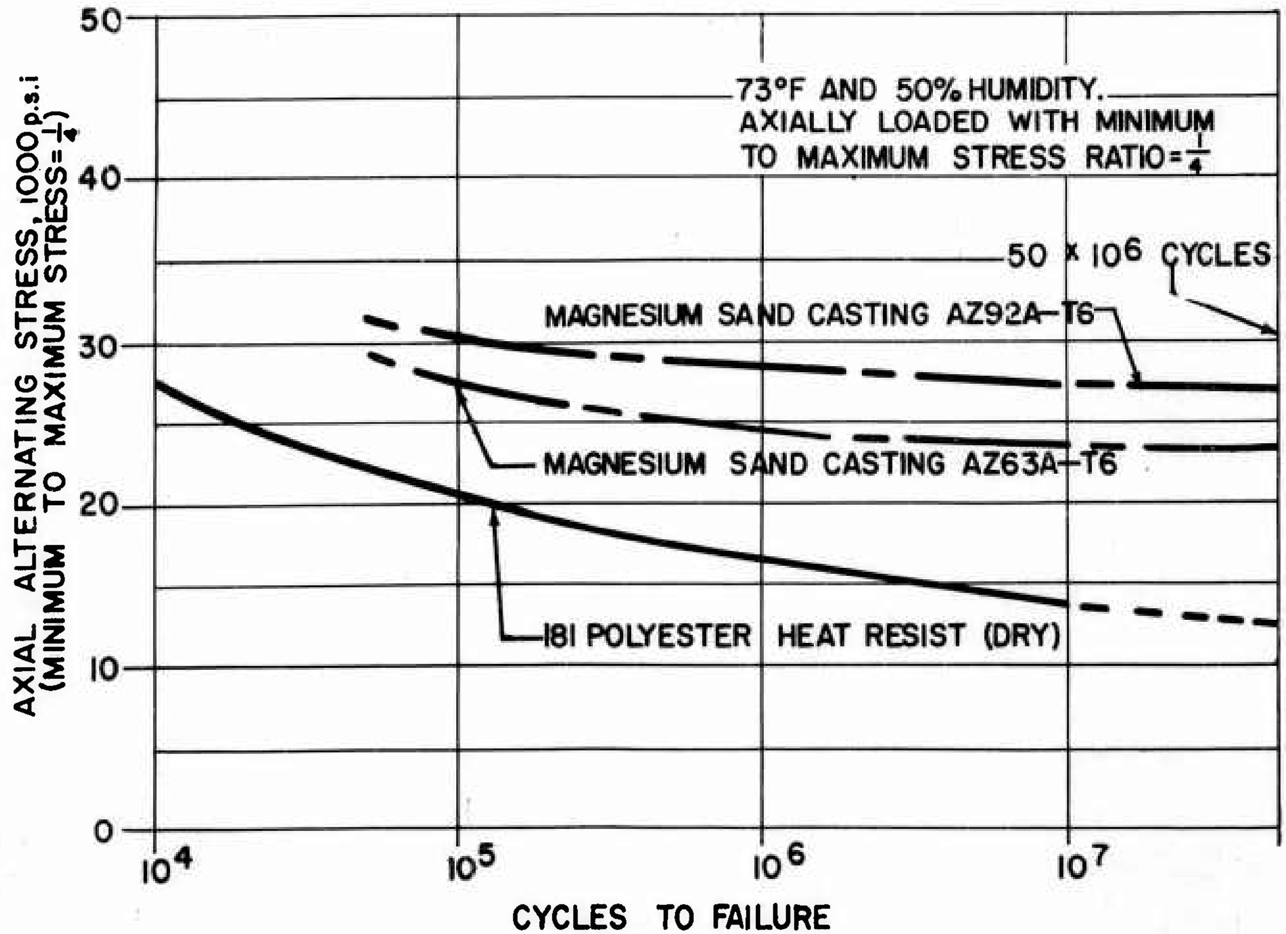


FIGURE 34. DIRECT STRESS FATIGUE CHARACTERISTICS – MAGNESIUM ALLOY VS. WOVEN GLASS FABRIC POLYESTER LAMINATES

PHYSICAL PROPERTIES OF MATERIALS

In comparing the physical properties of reinforced plastics and the light metal alloys, the two types of materials indicated a decided difference in the thermal expansion and thermal conductivity properties, as shown in columns 9 and 11 of Table   For the reinforced plastics, the coefficient of thermal expansion was considerably less than for magnesium and aluminum. Thus, the reinforced plastics have much better dimensional stability since they expand less for a given temperature rise.

The thermal conductivity values for reinforced plastics are small as compared to the light metal alloys, and this indicates that plastics are poor conductors or good insulators. Thus, it is apparent that the use of reinforced plastics where heat dissipation is important will require special design considerations. In the case of transmission housings, reinforced plastics may require a special heat exchanger.

Vibration damping characteristics are another physical property of importance in the selection of materials for transmission housings. It is desirable that the materials have good damping qualities, as vibration transmission is a serious problem in helicopters.

Very little information is available on the damping characteristics of the materials considered, but it is known that plastics have considerably better vibration damping characteristics than the light metal alloys. Quantitative information, however, is required to determine the order of magnitude of improvement different reinforced plastic materials would provide as compared to the light metal alloys.

COMPARISON OF MATERIAL PROPERTIES

An evaluation of the various mechanical-physical properties of the materials considered for transmission housings is shown in Table
This table shows the relative standing of each material for each property, and an over-all relative standing, assuming that weight, specific stiffness, thermal expansion, and vibration damping were of equal importance while creep strength and thermal conductivity were of lesser importance. This was done by assigning a value of 1 to 10 for the former properties and 1 to 5 for the latter. Thus, it might be said that creep strength and thermal conductivity are only half as important as the other properties by this procedure. The lowest number indicates the material with the best respective property, i.e., highest strength to weight ratio, highest specific stiffness, highest creep strength, lowest thermal expansion, highest thermal conductivity, and highest vibration damping characteristics. The respective standing of the other materials was determined by the ratio of the best mechanical or physical property to that of each material. The over-all standing was determined by summation of the standing for each property and grading each material accordingly, the smaller total indicating

TABLE 9
EVALUATION OF MATERIALS FOR TRANSMISSION HOUSINGS BASED ON COMPARATIVE MECHANICAL-PHYSICAL PROPERTIES

Material	Strength to Wt. Ratio		Specif. Stiff.	Creep Strength	Thermal Expans.	Thermal Cond.	Vibration Damping*	Sum		Overall
	Dry	Wet						Dry	Wet	
181 Polyester	4	3	4	2	6	5	5	26	26	4
181 Ht. Resist. Polyester	4	4*	5	2	4	5	5	25	25	4
181 Epoxy	3	3	4	2	4	5	5	23	23	3
181 Epoxide	3	2	4	2*	5	5	5	24	23	3
181 Ht. Resist. Epoxide	5	5*	5	2*	5*	5	5	27	27	4
Unwoven Parallel	2*	4	3	1	4	5	5	20	22	2
Unwoven Crossply	2	4	4	1	5	5	5	22	24	3
Unwoven Isotropic	2*	5	6	1	6	5	5	25	28	4
181 Phenolic	4	4*	4	2*	5	5	5	25	25	4
181 Silicone	6	6*	5	2	3	5	5	26	26	4
Parallel Asbestos Mat - Phenolic	2	2	3	1	2	5	5	18	18	1
Cross Lam. Asbestos Mat - Phenolic	3	3	3*	2*	3*	5	5	21	21	2
Fiberglass Mat - Polyester	10	10*	10	4	6*	5	5	40	40	5
181 CTL-91D Phenolic	4*	4*	4	1	5	5	5	24	24	3
AZ-63A-T6 Mag. Casting	2	2	2	5	10	2.5	10	31.5	31.5	5
AZ-92A-T6 Mag. Casting	2	2	2	5*	10	2.5	10	31.5	31.5	5
AZ-80A-T5 Mag. Casting	2*	2*	2	4	10	2.5	10	30.5	31.5	5
195-T6 Alum. Casting	10	8	2	2*	10	0	10	34	32	5
355-T6 Alum. Casting	10	8	2	2	10	0	10	34	32	5
356-T6 Alum. Casting	10	8	2	2	9	0	10	33	32	5
2024-T4 Alum. Alloy Plate	3	3	2	1*	10	0	10	26	26	4

*Estimated Standing - quantitative information is incomplete

the material with the better over-all mechanical-physical properties for this application.

By assigning values to the plastics considering their vibration dampening properties, it was noted that they are much more desirable materials than the light metal alloys. In this case, parallel laminated asbestos-phenolic mat was the leading plastic, followed by unwoven parallel epoxy laminate. Had the vibration damping properties of the plastics been neglected, then the metal alloys would have had superior ratings. It is believed, however, that in helicopter transmission housings, vibration effects cannot be neglected and that reinforced plastic materials are clearly superior to the light metal alloys.

Since the phenolic-asbestos mat laminate is one of the leading reinforced plastics considered in this evaluation and since this material requires a high laminating pressure (400 p.s.i.) during the molding process, it is important to investigate the effect of lowering the laminating pressure. Manufacturer's literature, Reference 84, indicates that the strength and elastic modulus values at a 50-p.s.i. laminating pressure are on the order of 75-85 percent of the values of 400 p.s.i. It is believed that the fatigue strength would be reduced similarly. Thus, by reducing the laminating pressure of phenolic-asbestos mat laminate to 50 p.s.i., it can be assumed that its properties would be almost equivalent to those of unwoven parallel epoxy laminate or cross laminated phenolic-asbestos mat molded at 400 p.s.i.

REINFORCED PLASTIC CONFIGURATIONS

The preceding evaluation of the mechanical-physical properties of various materials indicates that reinforced plastics are desirable materials for transmission housings. Typical transmission designs were therefore investigated to determine the feasibility of using these materials to replace conventional housing materials.

A review of current helicopter transmission designs indicates that they generally consist of the following primary components or subassemblies:

1. Input, main rotor, tail rotor, and auxiliary drive shafts.
2. Upper, intermediate, and lower transmission housings which have interconnecting mounting flanges, bearing mounting surfaces, and attaching lugs for mounting the transmission assembly to the helicopter frame.
3. Miscellaneous drive shaft housings and covers.
4. A lubricant distribution system.

The component housings are attached by bolted-flange connections, which in some cases must carry the main rotor loads to the adjacent housing and eventually to the fittings attaching the transmission to the helicopter framework. A housing design concept which can offer at least an equivalent structural efficiency for the reinforced plastic as the light metal alloys is required. Such a concept requires a housing configuration that will allow efficient orientation of the reinforcement. This is not believed to be practical with the present configuration. Therefore, the design concept would have to be changed to eliminate the intricate shapes that will not allow smooth flow and continuity of the plastic reinforcing materials. Designs using configurations involving surfaces of revolution are the simplest. Variations will add to the complexity of lay-up and molding and therefore increase the cost.

Preoriented preimpregnated material appears to be potentially feasible for this application. It permits excellent conformability while retaining high strength. See discussion of filament winding in Test Section.

In general, most of the reinforced plastic housings could be fabricated without any appreciable machining. This could be done by using precision mandrels and male or female forming dies. The primary machining requirement would be the drilling of holes for mounting flanges, etc.

Three fabrication processes are considered to be feasible for the reinforced plastic transmission housings. These are the vacuum bag, the autoclave, and the matched molds. Vacuum-bag molding results in the lowest strength and is considered to be feasible only where the strength requirements of the housing are not critical. The autoclave process appears to be the more feasible of the three because it allows high molding pressures (up to 200 p.s.i.) and offers good mechanical properties at moderate cost. The matched-mold die process offers a considerably higher laminating pressure range (400 p.s.i. and higher) and the best mechanical properties, but it involves a considerably larger investment in tool and die equipment.

The comparative unit costs of reinforced plastic versus cast light metal alloy transmission housings would depend primarily on such factors as tooling, raw materials, labor, inspection, and quality control. It is believed that the relative cost of the reinforced plastic housing, as compared to a cast light metal alloy housing, would vary from somewhat less to approximately equal depending upon the housing complexity and the number of units involved. However, considerable development cost would probably be incurred for a prototype design. A comprehensive cost analysis and comparison should include analysis of actual cost data from existing housings and the estimated cost of a reinforced plastic housing designed to meet the identical design loads and environmental criteria. Such an analysis has been beyond the scope of the present design study: more definitive materials information is required before the design and analysis of a specific configuration can be accomplished.

Following is a brief outline of cost factors of light metal and reinforced plastic designs.

	Light Metal Alloy Castings	Reinforced Plastic Laminates
Tooling Costs	Primary foundry equipment is already in existence. Essential tooling would be the sand molds or patterns, or permanent mold depending upon quantity.	Autoclave capable of 50 psi & 500°F necessary & is assumed available. Male mold & vacuum system would be primary tooling costs, which would probably be comparable to a casting permanent mold.
Raw Material	$1.00 - $2.00 per lb., depending upon complexity.	$2.00 to 2.35 per lb. (roll or sheet preimpregnated.)
Labor	Primary labor requirements are for precision machining of casting for flange surfaces and bearing bores.	Primary labor requirements are for lay-up or reinforced plastic & operation of molding equipment. Machining would be minimized to drilling bolt holes, etc.
Inspection & Quality Control	X-ray of all castings required & each housing would require close quality control of machined dimensions on each housing.	Continual process inspection. Dimensional quality control limited to initial quantities & spot check of succeeding quantities.

The pertinent advantages and disadvantages of reinforced plastic transmission housings are listed.

Advantages

1. Fatigue strength to weight ratio equivalent to or possibly superior to magnesium castings and considerably superior to aluminum castings.

2. Creep strength superior to or equivalent to aluminum and considerably superior to magnesium.

3. Thermal expansion approximately one-half that of magnesium or aluminum.

4. Predicted superior vibration damping characteristics.

5. Possibly equivalent or slightly less production unit cost based on simplified design and minimizing machining requirements.

Disadvantages

1. Less rigidity or stiffness under load than aluminum or magnesium.

2. Considerably lower thermal conductivity than aluminum or magnesium (less ability to dissipate heat).

3. Difficult fabrication problems in housings with complex shapes and contours.

4. Research and development required for optimized applications.

CONCLUSIONS

As a result of this investigation of the application of reinforced plastic laminates to transmission housings, the following conclusions are drawn:

1. The mechanical-physical properties indicate that reinforced plastics are superior to light metal alloys if minimum thermal expansion and adequate vibration damping are essential design criteria. (This conclusion is based on assumed comparative damping characteristics for reinforced plastics and light metal alloys.) Otherwise, reinforced plastics and light metal alloys have almost equivalent mechanical-physical properties. The light metal alloys have the advantage of well developed manufacturing techniques.

2. The reinforced plastic which exhibits the most desirable mechanical-physical properties is phenolic-asbestos mat laminate cured at a molding pressure of 400 p.s.i.

3. The reinforced plastic laminates which are feasible for general transmission housing applications, from the standpoint of materials suitability, are as follows:

   a. Phenolic-asbestos mat laminates.

   b. Unwoven epoxy laminates (parallel and crossply).

   c. 181 woven epoxy laminates.

4. It will be necessary to develop new design approaches for feasible reinforced plastic transmission housings. Simplicity of design is essential for feasible fabrication processes.

5. It will be necessary to develop certain fabrication techniques for feasible applications of reinforced plastics as transmission housings. Some of the anticipated problem areas are:

    a. Development of lay-up techniques and molding facilities (mold, pressure bags, etc.) to accomplish the intricate contours, flanges, and reinforcing ribs necessary in transmission housings to the degree consistent with accomplishment under Item 4 above.

    b. Determination of those transmission housing components (bosses, flanges, reinforcing ribs) which may be preformed and molded in place with the main portion of the housing or bonded to the housing after it is fabricated. Also, determination of the method of incorporating the lubricant distribution system into a reinforced plastic housing.

    c. Determination of the tolerance requirements of housing precision surfaces and determination of those surfaces which will require machining. It may be possible to locate preformed or prefabricated bosses with adequate precision to eliminate machining after molding the housing.

    d. Investigation of various quality control techniques.

6. The feasibility of reinforced plastic transmission housings is indicated to a degree justifying further effort, in order to achieve the following potential benefits.

    a. Improved vibration damping - reduced noise levels.

    b. Increased fatigue and creep strengths - longer service life.

    c. Reduced thermal expansion - longer service life.

RECOMMENDATIONS

As a result of this study of the feasibility of reinforced plastic transmission housings, it is recommended that further investigation, design and development be accomplished in accordance with the following sequence. Each part of the program should be justified on the basis of favorable results in the preceding part of the program.

1. Conduct vibration damping tests to determine the relative damping characteristics of representative reinforced plastics and the metal alloys. These tests should determine the damping characteristics in the longitudinal and transverse directions (direct and flexural stresses) and should cover a frequency range of approximately 10 cps through the essential audio frequencies of approximately 10,000 cps maximum.

2. Conduct an analysis of the vibration and noise problem in transmission housings to determine the significance of housing material damping characteristics.

3. Conduct tests to confirm the mechanical-physical properties of phenolic-asbestos mat laminates molded at lower laminating pressures of 50-100 psi. This shall include direct stress fatigue tests similar to those conducted in References 16 and 17.

4. Select a transmission housing design from one of the more advanced Army helicopters which shows promise of continued use in the Army inventory for the next 5- to 10- year period. Design a complete replacement housing using the most advantageous reinforced plastic materials and using the identical design loads and environmental criteria. Drawings should be sufficiently complete to build a prototype housing.

5. Conduct a cost analysis of the above reinforced plastic design and compare with the cost of the existing design.

6. Analyze the results of (1) through (5) and determine the feasibility of initiating a fabrication and development/qualification test program. Would it be economically feasible to develop a replacement reinforced plastic housing? If not, would it be worthwhile to use reinforced plastics in housing for new aircraft designs? This will require re-evaluation of the gain in overall mechanical-physical properties, possible gain in manufacturing advantages, and comparative logistics requirements.

7. If indicated by the results of (6), fabricate a prototype housing and conduct necessary qualification tests to finalize the design and generate a set of production drawings and specifications.

8. Prepare a final report to present the results of steps (1) through (7).

EMPENNAGE DESIGN STUDY

Army aircraft empennage configurations are considered to be in one broad category of design studies due to their basic similarity of size, shape and general requirements. All are relatively small streamlined surfaces, whether fixed or movable, and are designed primarily by airloads. The vertical fins of light and medium-weight observation and utility type helicopters are usually fixed surfaces mounting tail rotors for directional control. Horizontal stabilizers may be fixed or controllable depending upon the helicopter configuration and stability requirement. Controllable surfaces may be all-movable, or combined fixed and hinged surfaces. Light fixed-wing aircraft usually are configured in the conventional manner, with the vertical tail consisting of fixed fin and movable rudder and the horizontal tail consisting of fixed stabilizer and movable elevator. Future Army aircraft of unconventional configuration are envisioned. However, the majority of unconventional designs employ conventional tail surfaces, and the results of this study will be applicable in those cases.

Generally speaking, industry practice has been to design and fabricate tail surfaces of conventional sheet-metal-formed parts and skins. No applications of reinforced plastic to production-type aircraft empennages are known other than the currently projected Piper PA-29 development and the H-43 outboard vertical stabilizers. Related applications have been made in the KC-97 refueling boom trim vanes, in missile fins, and in several experimental aircraft.

Preliminary analysis of existing Army aircraft empennages indicates that the basic significant difference in the various types of surfaces is the structural complexity. Since there is considerable potential variation in the degree of feasibility of very simple structural configurations and more complex designs, it was apparent that at least two general types of surfaces should be studied. The light helicopter all-movable horizontal stabilizer is typical of the more simple structures. The light fixed-wing aircraft horizontal tail, with its fixed stabilizer, movable elevator and trim tab, is representative of the more complex structural problems. Studies of thesd two types are indicative of the majority of Army aircraft applications and permit detailed examination of both the helicopter and fixed-wing aircraft design requirements. (Design criteria considered applicable to control surfaces and empennages are contained in References 7, 21, 42, 46, 49, 50, and 51).

In order to provide a basis for comparative evaluation of reinforced plastics and conventional design, existing aircraft configurations were chosen for study. The HU-1 horizontal tail was selected because it represents somewhat of an "average" in size and simplicity and also because the aircraft series is representative of current and near-future medium size helicopter procurement. The L-23 horizontal tail was chosen as the second example for study since it is typical of light fixed-wing aircraft and other similar design problems and represents an anticipated "average" in complexity. These studies are also applicable to vertical tails, ailerons, and flaps.

HU-1 ELEVATOR

The present HU-1 elevator is of all-metal construction. It consists of a tube acting as the main spar to which sheet metal ribs are attached and the assembly covered with metal skin. The two elevators, one on either side of the tail boom, are supported by bearings at the side of the tail boom. The torque tubes are joined together at the centerline of the tail boom with a control horn fitting.

Several different reinforced plastic materials and methods of construction were investigated. Four of the most feasible configurations representing different approaches are summarized.

All four configurations are shown in Figure 35. The basic design concept is the same as for the metal elevator. The torque tube which also acts as the spar is retained. Each configuration uses a rectangular tube for the spar, tapering to a cylindrical section for the bearing area. The variation in the different configurations is in the construction of the airfoil shape. The torque tube as shown is constructed of fiberglass. Although this is feasible, some cost saving could result if the tubes were made of metal and the bearing size changed to eliminate the necessity of tapering the tube.

Configuration I is a solid laminate preformed shell bonded to the tube and together at the trailing edge. Closure ribs at the ends are bonded in place.

Configuration II is similar to I except that the shell is much thinner and is stabilized with rigid foamed-in-place foam after the shell is bonded to the tube.

Configuration III consists of two sandwich panels forming the upper and lower contoured surfaces. These preformed panels with aluminum honeycomb core and fiberglass faces are spliced together at the leading and trailing edges and bonded to the torque tube.

Configuration IV is similar to III except that the contoured surfaces are fabricated from an integrally woven fluted core panel called "Raypan". This is a relatively new development by Raymond Development Industries, Inc.

A cost and weight comparison of these configurations is given in Table 10.

Several other configurations were investigated to a lesser extent and are not summarized in detail. These included sandwich construction with full-depth honeycomb core. This gives higher strength, but weight and cost would also be greater. This method of construction would be appropriate for surfaces that are more highly loaded. Thin skin supported by full-depth ribs is another feasible method of construction. This type would be lighter than the others, but it would be less rugged and the cost would not be any less.

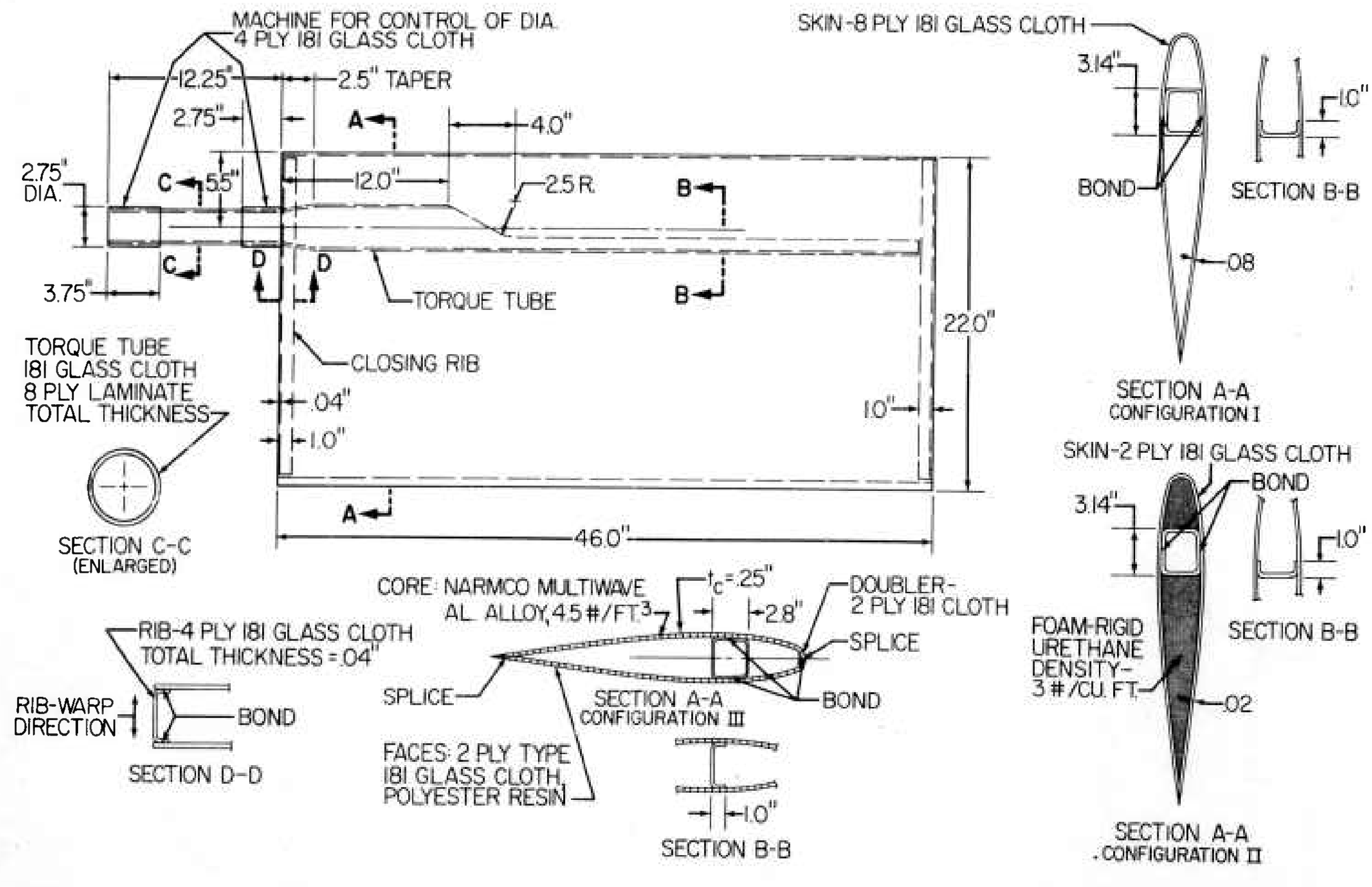


FIGURE 35. REINFORCED PLASTIC DESIGNS OF HU-1 ELEVATOR

The laminates for each configuration can be fabricated with Type 181 cloth preimpregnated with polyester resin. Female molds should be used to insure a smooth outside surface. Vacuum-bag molding will provide sufficient pressure. The tube can either be laminated or filament wound. Strength characteristics of filament-wound tubes are given in the Test Section.

TABLE 10
HU-1 ELEVATOR - WEIGHT AND COST COMPARISON

Configuration	Weight Pounds	Unit Cost	
		Quantity of 10	Quantity of 100
I Solid Laminate Skin	13.9	$1020	$440
II Foam Stabilized Skin	8.1	1050	395
III Honeycomb Sandwich	9.6	1000	455
IV Integrally Woven Fluted Core	13.5	425	350
Existing Metal	13.5	$315	

*Cost of spares as listed in Federal Stock Catalog.

EVALUATION

This study indicates that fiberglass reinforced plastic is a feasible material for use in the small control surfaces. Several different methods of fabrication were investigated and four were found to result in equal or less weight than the existing metal structure. The cost estimates as presented are slightly higher than the cost of the metal elevator as listed in the Federal Stock Catalog. These estimates are believed to be conservative; with newer developments in tooling material and methods with a concentrated cost reduction study, the cost can be reduced below the metal cost for quantity production. The design of the metal component is considered to be quite simple, and further optimization would not significantly reduce cost or weight. An optimized reinforced plastic design should result in reduced weight and cost.

Other advantages of reinforced plastic for this application are as significant as the possible weight reduction. The surface can be made aerodynamically more efficient because of the smooth molded surface and skin that does not buckle under load. Faired shapes can be obtained easily with plastics that enhance the appearance. A durable finish and color can be molded into the part to minimize maintenance. It is understood that on some helicopters the exhaust from the turbine engine is directed in such a manner that the paint is burned off the control surfaces. Heat-resistant plastic finishes should alleviate this problem.

The advantages and disadvantage of the use of reinforced plastics for small control surfaces can be summarized as follows:

Advantages

1. Lighter Weight
2. Greater Aerodynamic Efficiency
3. More Durable Finish
4. Durable Surface
5. Easily Repaired
6. Radar Transparent
7. Lower Cost Potential

Disadvantage

Requires development to obtain optimum design and fabrication procedure.

L-23 HORIZONTAL TAIL

The L-23D horizontal tail, unlike the all-movable HU-1A elevator, is composed of three distinct parts: a fixed surface or stabilizer which mounts rigidly to the fuselage, a movable surface or elevator which is hinged to the aft edge of the stabilizer and is approximately equal in size to the stabilizer, and a trim tab which is hinged to the rear of the elevator and is about ten percent the size of the elevator. The structure is much more complex in design, requiring considerably more tooling and fabrication and resulting in a more costly item.

The function of the horizontal tail on a conventional aircraft differs from the helicopter in that the airplane depends completely upon the tail for flight equilibrim. Equilibrium is accomplished by having a tail which is capable of balancing tail-off unbalanced pitching moments. A balancing load of the proper magnitude and direction is produced by the elevator's forcing the tail to the proper attitude. The required balancing loads vary for the different flight operating conditions imposed upon the aircraft.

A variety of wing-type structures are used in subsonic light airplane design. However, they may be categorized into two major groups: (1) those that have bending material around the section periphery and (2) those that contain most of the bending material in spar caps. Generally, they all contain a torsion box, formed by a front and rear spar and the cover material between them, as the primary structure. The structure ahead of and behind the torque-box is usually just an aerodynamic covering with adequate strength to support directly applied airloads and to transmit these loads to the primary structure. Normally, the leading and trailing edge structure is not used to carry any primary shear, bending or torsion loads. The wing covering is designed primarily to carry shear imposed by torsion and some shear due to chordwise load and shear lag. Bending material for light airplanes, which are lightly loaded, is designed into the spar caps and usually has high stress allowables capable of carrying all the bending loads. Often this is the most efficient approach, rather than adding material to the skin and stringers, which are limited to lower allowable crippling stresses than the spar caps even though they are further from the section neutral axis. As support to the bending material, which carries compressive loads and tends to buckle as a column, ribs are used in normal construction practice. The number of ribs employed is usually determined by the requirement to make the column allowable equal to the crippling allowable.

The aluminum alloy L-23 horizontal stabilizer is of two-spar construction. The covers are stiffened by chordwise beads and by ribs at the elevator hinge locations. Thin sheet skin supported by closely spaced ribs makes up the leading edge. The stabilizer is attached to two stub spars at the side of the fuselage by multiple bolt shear splices.

The elevator has a single spar. The leading edge and trailing edge are thin sheet supported by ribs. Additional stiffness of the trailing edge is obtained by internal beads. Magnesium alloy is used for the skin, spar, and small ribs. The hinge ribs are aluminum alloy. A torque tube attached to the inboard end actuates the elevator.

The tab is constructed of two contoured sheets of magnesium alloy riveted together at the leading and trailing edges. It is attached to the elevator by a piano hinge.

All three major components of the L-23 horizontal tail were investigated for the applicability of reinforced plastic construction. Load data used for preliminary analysis and design study were based on Section 3.215, Balancing Loads, of Reference 21.

Design consideration was given to the various methods of fabrication investigated for the HU-1A elevator; namely, foamed core, plain plastic laminates and honeycomb sandwich. For lifting surfaces as large as those normally required for airplanes, it was found (1) that a foam-filled structure had the disadvantage of excessive weight due to the large foam volume within the airfoil section and high density requirement; it also required either a moderately heavy torque-box structure for adequate transfer of combined aerodynamic bending and torsional loadings into the fuselage or a heavy torque-tube, as the case may be: and (2) that for monocoque structure, laminated skins become excessively heavy due to the large panel sizes resulting from compressive buckling requirements of the unsupported skin. Hence, the sandwich provided a means for eliminating the heavy spar or tube requirement for the stabilizer and elevator by using the entire airfoil section for transmitting torque. Shear between the upper and lower surfaces is transferred by a spar of snadwich or solid laminate construction. Through use of honeycomb core, which can be either nylon phenolic or aluminum, sufficient panel rigidity is provided and the volume of material required for skin stabilization is reduced from the thick laminate approach. A sandwich having a foam core can also be used, but it would have less strength and stiffness than one using a honeycomb core of the same weight.

When reinforced plastics are used for this type of structure, the configuration should be such that the loads are distributed rather than having concentrated load paths as in the present metal construction that uses two spars to carry the primary bending and shear loads. The most efficient reinforced plastic design for the stabilizer is one using the entire upper and lower surfaces to carry bending and torsional loads. This configuration would require that the box structure be continuous through the fuselage. It could not be substituted for the present metal stabilizer without extensive redesign and modification to the aft fuselage.

A sketch of a feasible reinforced plastic configuration for the stabilizer is shown in Figure 36. The box structure consists of two spars and the upper and lower covers. These components are fabricated separately and bonded together to form the box structure. The leading edge, trailing edge fairing and tips are attached separately by bonding or with mechanical attachments. The spars are laminated channels that can be fabricated from woven preimpregnated fabric in a female mold. Sandwich construction can also be used for the spars if desired. The covers are of honeycomb sandwich with faces of Type 181 glass cloth and epoxy resin.

Each sandwich component can be fabricated in the same manner in a female mold with a single-cycle cure process. The outer face is laminated as a wet lay-up using Type 181 cloth and epoxy resin. The core material, preshaped as necessary, is then placed in position. The required local reinforcements and border members are laminated along with the sandwich, or they may be premolded and placed in position. After the core and border members are placed in position, the inner face is laid up using the same material as for the outer face. The complete assembly is then cured in an autoclave or in an oven with vacuum-bag pressure.

The leading edge, trailing edge, ribs and tips are solid laminates and can be fabricated using epoxy preimpregnated Type 181 cloth in a female mold, using vacuum bags and an oven or an autoclave to cure.

Figures 37 and 38 show feasible methods of construction for the elevator and trim tab. The two-piece laminated shell is made in female molds, bonded together and then filled with foamed-in-place urethane foam. The foaming operation would require the use of a fixture to retain the shell under the foaming pressure.

Table 11 shows a weight and cost comparison of the reinforced plastic components and the present metal components.

TABLE 11
L-23D HORIZONTAL TAIL - WEIGHT AND COST COMPARISON

Component	Weight (Semi-span) (pounds)	Unit Cost (dollars) Quantity of 10		Quantity of 100
Plastic Stabilizer	46.8	2150		720
Metal Stabilizer	32.0		320*	
Plastic Elevator Structure	10.4	1670		490
Plastic Elevator Balance Wt.	8.5			
Metal Elevator Structure	8.1	—	315*	—
Metal Elevator Bal. Wt.	5.9		—	
Plastic Trim Tab	1.4	995		270
Metal Trim Tab	1.2		40*	

*Costs of spares as listed in Federal Stock Catalog. These values appear to be much too low. No other source is available for check.

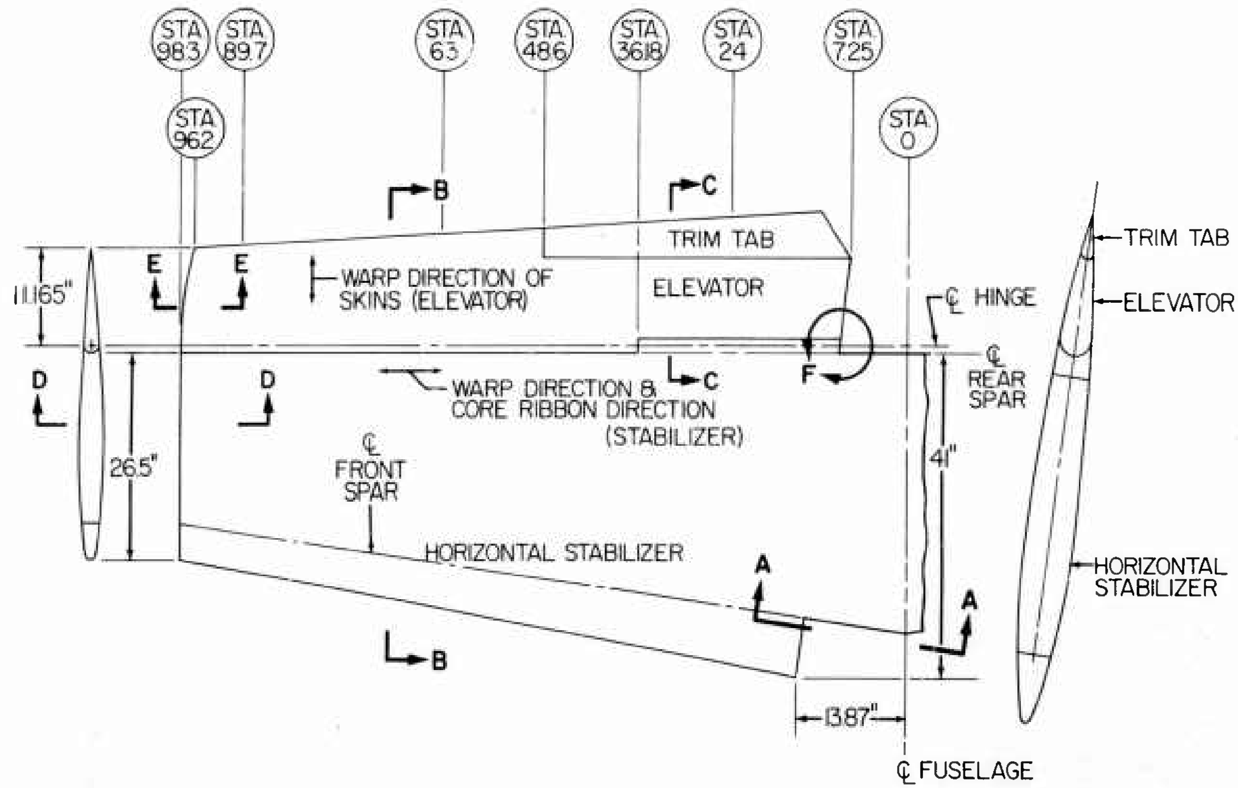


FIGURE 36. L-23 HORIZONTAL TAIL

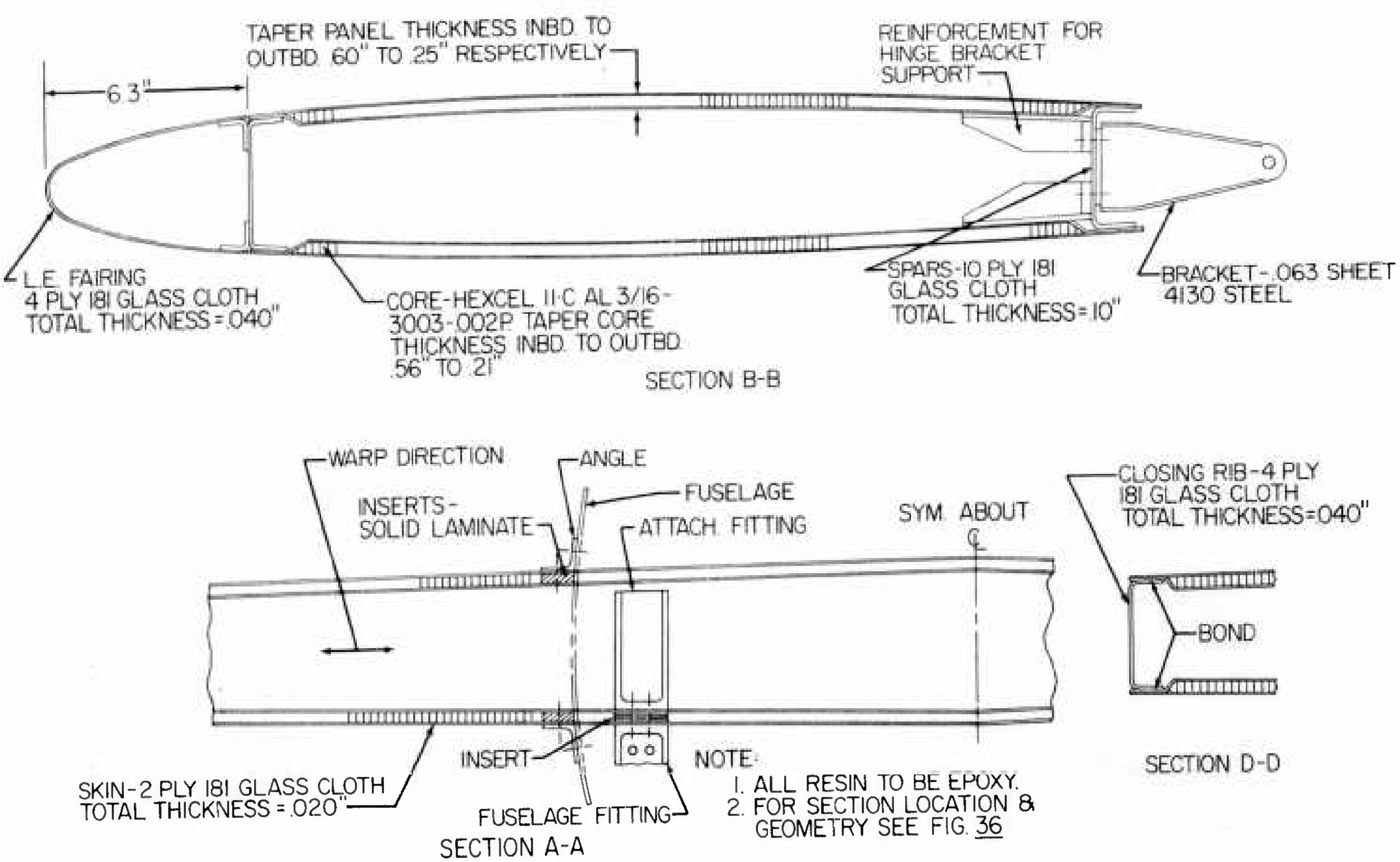


FIGURE 37. L-23 HORIZONTAL STABILIZER OF REINFORCED PLASTIC

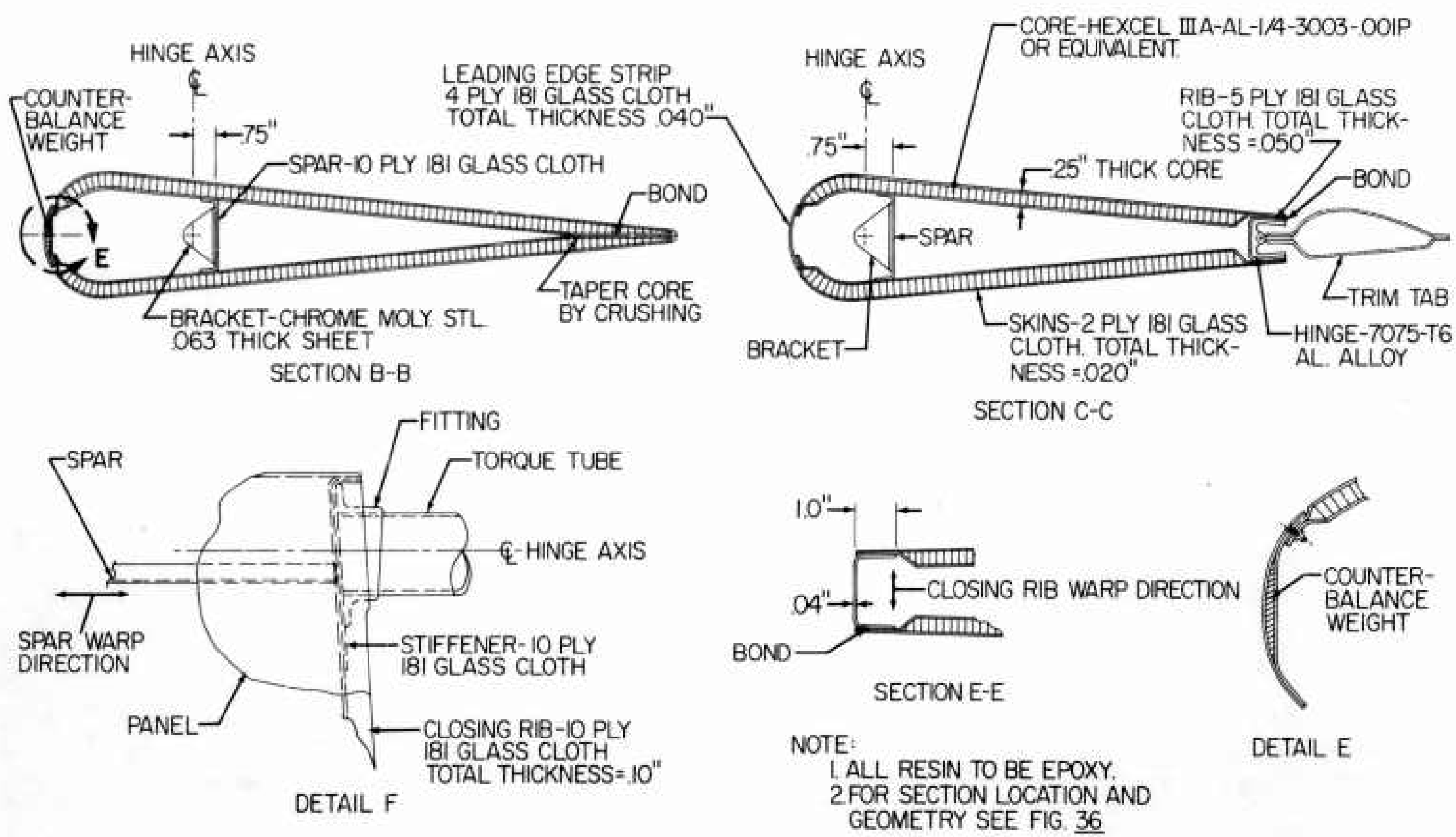


FIGURE 38. L-23 ELEVATOR OF REINFORCED PLASTIC

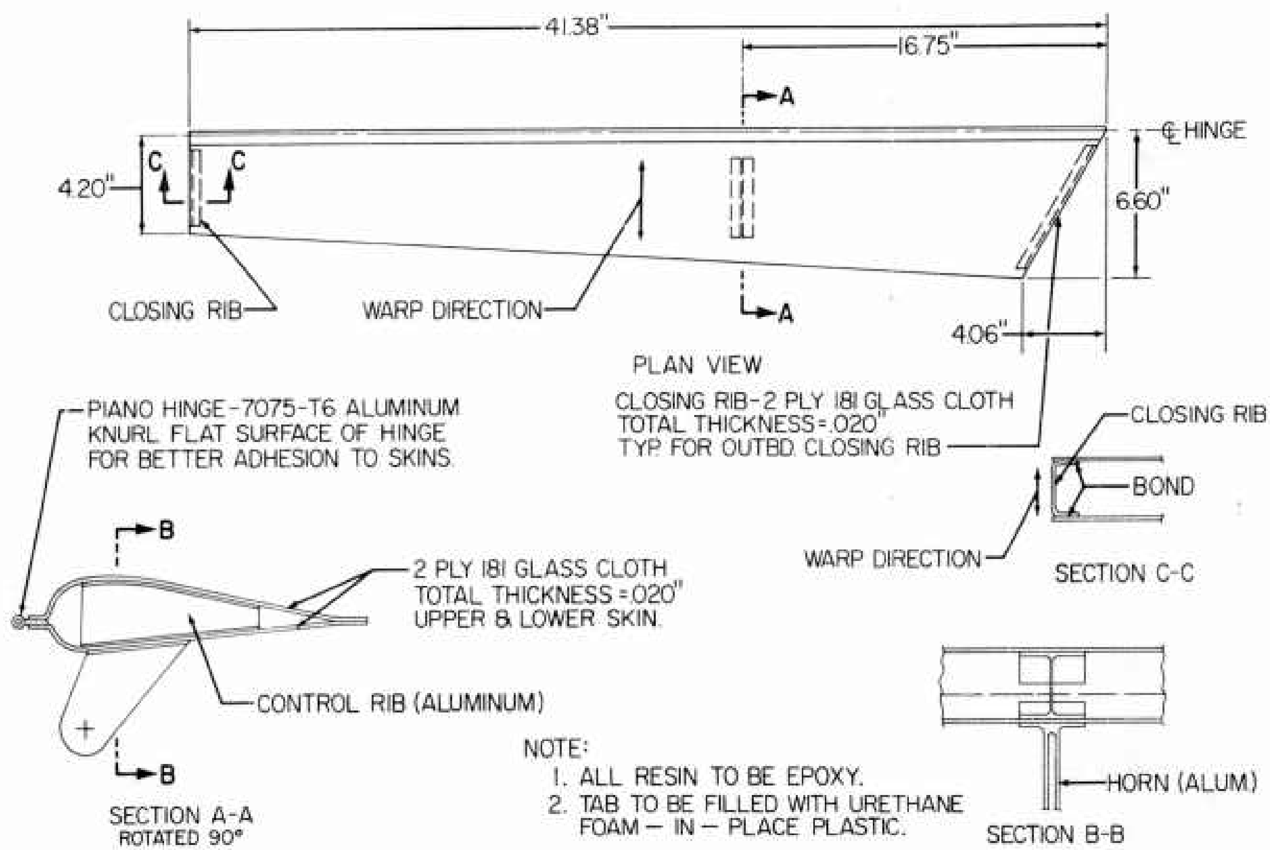


FIGURE 39. L-23 ELEVATOR TRIM TAB OF REINFORCED PLASTIC

Many of the advantages cited in favor of plastics over metal construction in the HU-1A discussion are equally applicable to the L-23 horizontal tail case. These factors need not be repeated; however, some additional comments regarding the use of sandwich construction is in order.

Sandwich construction has many structural advantages over solid sheet metal construction, among which is the fact that the thin gauge facings can be designed to very high allowable compressive stresses since they cannot buckle until the entire section fails. Light weight rigid panels can be fabricated using low-density core materials which, when used in conjunction with thin facings, permit the facings to be nonbuckling under shear loading up to their ultimate shear stresses.

Some of the disadvantages of sandwich construction lie in the fact that (1) the core material must be precision machined to provide an adequate face bonding surface; (2) forming the sandwich to complex contours is difficult for most core materials - NARMCO Multiwave sandwich eliminated this problem for some core materials; and (3) splicing of honeycomb sections and installation of required fittings increase weight, are difficult to accommodate and are costly operations.

Evaluation

Analysis shows that presently available materials and fabrication techniques can result in structurally adequate reinforced plastic tail surfaces. Although a preliminary weight estimate indicates a 40 percent increase in weight, it is believed that design optimization will reduce this difference. The computed deflection for the plastic stabilizer is lower than would be expected for this material. This is undoubtedly due to the low stress at which the plastic is working (3000 to 6000 p.s.i. at limit load), even though the sandwich faces are quite thin. The cost data available for the metal components are not believed to be reliable enough for any true comparison. In summary, it can be said that the reinforced plastic components are competitive with the metal components as to weight, performance, and cost, but no outstanding advantage for either material is apparent.

It is concluded that reinforced plastics are feasible materials for the empennage of light fixed-wing Army aircraft. No unusual design and fabrication problems are visualized. However, considerable development and tests would be required to produce a prototype structure. The major advantages and disadvantages of reinforced plastic tail surfaces are:

Advantages

1. Improved aerodynamic efficiency due to smooth molded surface and lack of wrinkles under load.

2. Improved fatigue characteristics.

3. Radar transparent

4. More durable surface and finish.

Disadvantages

1. Development of optimum design techniques and fabrication processes required.

2. Probably heavier.

WING DESIGN STUDY

The potential quantity of Army aircraft in the fixed-wing category is not nearly so great as rotary-wing aircraft. Nevertheless, real improvements in fixed-wing cost, weight and performance are just as much desired. Significant improvements will, of course, greatly enhance the relative importance of this type of aircraft. The fixed-wing area of interest is considered to be in the subsonic light- to medium-weight category. Future configurations of fixed wings will be structurally similar to current modern aircraft.

High strength to weight reinforced plastics offer potential benefits here as in other structural applications; however, the aeroelastic and structural dynamic requairements for aircraft wings present significant problem areas which have received little, if any, attention in the past. Increased aerodynamic smoothness obtainable with reinforced plastics can result in significant gains in aircraft performance. These advantages are discussed in another section of this report.

As early as 1947, an Air Force program under the AMC Aircraft Laboratory investigated glass reinforced plastic wings and concluded that this material application was feasible. Current wing development for the all-plastic Piper PA-29 aircraft is the only known projected production application. No specific information was available on this aircraft. Work at Mississippi State University on their all-plastic research aircraft is of interest; it has been directed more toward aerodynamic than structural research and development.

Potential applications of reinforced plastic technology to wings do not fall into a specific type of design problem area, nor can they be generalized for a typical configuration study. A detailed wing design anlaysis is quite time consuming, involving unsymmetrical bending analysis and multiple cell shear analysis for several combinations of loading. However, for light aircraft in particular, a structure designed to take maximum vertical bending moments with some regard for other loadings will approximate very closely the required design for all loadings. Therefore, the wing of a typical light fixed-wing aircraft, the L-23, was selected for basic evaluation of the feasibility of using reinforced plastics. In this analysis, the condition of maximum beam bending moment was selected from Beech Aircraft Corporation Report No. 64-701, "Wing Stress Analysis - Outer Panel", Model L-23D (Reference 11).

The existing aluminum alloy wing is a two-spar configuration with conventional skin and stringer panels and rib supports. It incorporates a four-point attachment of the outer wing panel to the center wing. This two-spar configuration is maintained in the reinforced plastic design; however, the four-point splice is considered to be prohibitively inefficient and uneconomical in a plastic design, and therefore a continuous splice is assumed. This assumption is appropriate since the objective of this study is to evaluate the effective use of plastics in future designs, not necessarily the interchangeable replacement of components in existing designs.

Sandwich construction is considered to be the most feasible method of using reinforced plastics in this application. The nose structure is assumed to be effective in bending and shear, resulting in a two-cell configuration except at the inboard end. Since the nose section is discontinuous inboard of station 99, it is not effective in bending at this station. The analysis is based on the use of Type 181 glass cloth impregnated with epoxy resin for the faces and aluminum honeycomb core. The necessary material to resist the bending loads was determined and compared with the metal wing. The analysis was accomplished by automatic computation of section properties and bending stresses at several stations along the spars. As stated previously, a complete wing design would require more effort than could be expended in this program. In this simplified approach the material required and other properties are compared to a few stations. Such a comparison will indicate whether the material has potential and it is considered appropriate. The results of the analysis to determine the panel buckling stresses for the sandwich panels are summarized in Table 12.

TABLE 12
SUMMARY OF L-23 WING DATA

	Sta. 99	Sta. 141	Sta. 198	Tip
Reinforced Plastic				
Core thickness (between spars) - in.	1.41	1.19	.70	.20
Core thickness (leading edge) - in.	-	.19	.20	.20
No. of plies - outer face	5	3	3	3
No. of plies - inner face	4	3	2	2
EI - lb. in^2.	678×10^6	326×10^6	127.5×10^6	-
F_{cr} - p.s.i.	34,800	32,800	16,800	-
F_c - max. p.s.i.	31,754	23,932	11,607	-
Metal				
EI - lb. in^2.	-	951×10^6	280×10^6	-

Although some stiffness could be added to the plastic wing by a redistribution of area across a section, it is quite obvious that the plastic wing

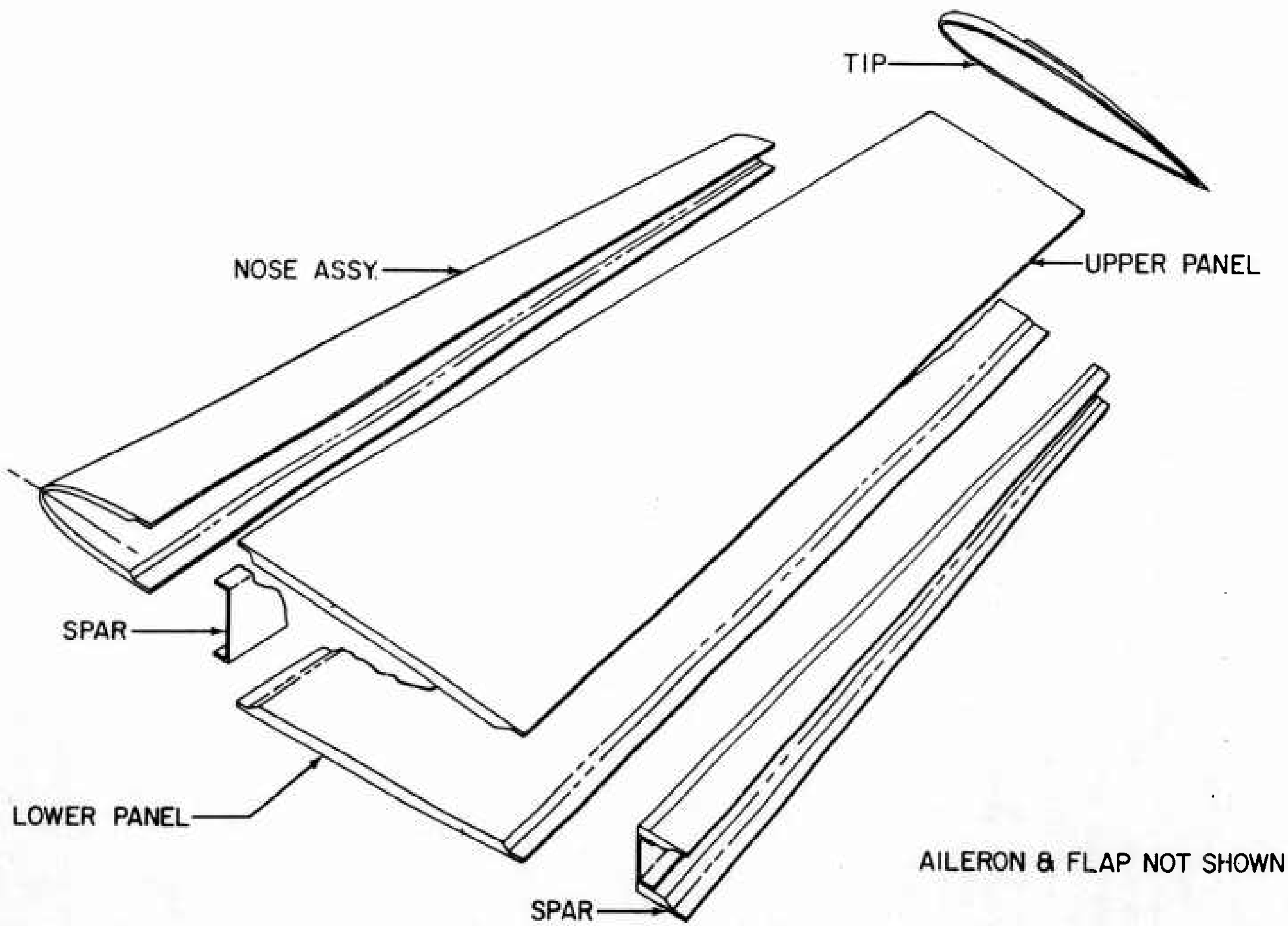


FIGURE 40. CONCEPT OF CONSTRUCTION OF L-23 WING OF REINFORCED PLASTIC SANDWICH DESIGN

will be appreciably more flexible than the aluminum wing when designed for strength alone.

Under static loading, the effect of the increased flexibility is negligible. Under dynamic loading, the effect is variable. For example, the more flexible wing would reduce the load factor at the center of gravity of the aircraft resulting from atmospheric turbulence. However, under certain gust profiles, the wing bending stresses could be magnified to a greater extent with the more flexible wing.

The greater flexibility would reduce the critical flutter speed while the increased damping of plastic structures would have the opposite effect. An appreciable amount of quantitative data on damping and a rather extensive analysis are required to evaluate the net effect of the use of plastics on flutter speed. However, for the normal light aircraft configuration, wing flutter is generally not considered to be a major consideration.

The following method of fabrication is considered to be the most appropriate. The primary structure is made up of five major subassemblies. These five members, consisting of the spars, upper cover, lower cover, and leading edge, can all be fabricated in the same manner with a single cure cycle. In order to obtain maximum smoothness on the external surface, female molds should be used. The outer face is first laminated as a wet lay-up using Type 181 cloth and epoxy resin. Pretrimmed and, if necessary, preformed core material is then placed on the outer face. No additional adhesive is necessary when epoxy resin is used. Necessary border members and local reinforcements are laminated along with the sandwich or are premolded and placed in position. The procedure to use is dependent on the requirements of the individual part. The inner face is laminated on the core in the same manner as the outer face. Pressure is applied with a vacuum bag or autoclave and the component cured.

Higher strength can be obtained by a three-step process in which the face laminates are premolded and then bonded to the core. This requires more tooling, operations, and labor results in a significant increase in cost. The test results show that sufficient strength can be obtained by the single cure method. The extra expense required to obtain the additional strength is not justified for these components.

The nose section can be installed in several ways. It could of course be bonded in place, making it a fixed installation and providing maximum aerodynamic cleanness. If it were desirable to have the nose section removable, it could be installed with screws. Under some circumstances it might also be desirable to attach it with a piano-hinge type joint.

Provisions must be made for accommodating hinges and actuators for flaps and ailerons. It will be necessary to provide structure to transfer hinge loads into the wing. Formers could be mounted to the wing blanket prior to wing final assembly and clearance holes provided in the rear spar to permit hinge brackets to extend aft. In some cases it may be possible to design the structure aft of the rear spar such that it can support the

hinge brackets and distribute these loads sufficiently for them to be transferred through the normal attachment of the trailing edge to the main wing structure. These additional parts would be molded individually and bonded or attached with mechanical fasteners to the basic structure.

EVALUATION

Reinforced plastics offer some very real advantages in application to aircraft wing structure. Probably the greatest of these is the performance gain due to aerodynamic cleanness. The aerodynamic qualities of reinforced plastics are discussed in another section of this report.

A second important advantage of plastics is the apparent improved fatigue characteristics of this material. Several things contribute to this condition, including:

1. Better basic material fatigue properties
2. Less notch sensitivity in fatigue
3. Better damping characteristics
4. Fewer and less severe discontinuities inherent in structures designed for reinforced plastics.

A wing designed for plastics contains a minimum number of ribs, leaving more clear area for such internal installations as fuel tanks.

This analysis indicates that a wing designed in plastics has equivalent or less weight than one in metal; however, the plastic wing is more flexible. For the relatively low-speed and lightweight aircraft envisioned for this application, this increased flexibility is not considered to be significant.

Due to the preliminary and general nature of this investigation, it is not practical to make a quantitative cost analysis. It is believed that the reinforced plastic wing can be competitive with conventional metal costwise. However, considerable development is required to optimize the design and the fabrication process.

It is concluded that reinforced plastics are feasible materials for wings of light fixed-wing Army aircraft. No unusual design and fabrication problems are visualized; however, considerable development and test would be required to produce a prototype structure. The major advantages and disadvantages of reinforced plastic wings are:

Advantages

1. Improved aerodynamic efficiency due to smooth molded surface. The contour is maintained because the skin does not buckle under load.

2. Improved fatigue characteristics

3. Radar transparency.

4. Potential lower cost and weight.

Disadvantages

1. Development of optimum design techniques and fabrication processes required.

2. Aeroelastic and structural dynamic effects require investigation.

FUEL TANK DESIGN STUDY

The basic requirements for aircraft and helicopter fuel tanks are satisfactory service life, compatibility with the fluids for which they are intended, high fatigue strength-to-weight ratio, good corrosion resistance, and low cost. Reinforced plastics offer several potential advantages in these areas.

Fuel tanks and other liquid containers such as oil tanks used in Army aircraft and helicopters may be classified into one of three configuration groupings: flexible tanks, integral tanks and individual rigid tanks. The first two groups are excluded from this discussion since they depend on the surrounding structure for their strength and are normally not designed as rigid tanks but as bulkheads, spars, or frames with additional loads due to fuel. The more prevalent group is the irregularly shaped, box-like rigid structure designed to fit in the available space in a specific location in the aircraft. This type of tank is usually made from one of the weldable aluminum alloys such as 5052-0, which also has good formability. Another configuration includes those tanks formed as a symmetrical body of revolution such as the streamlined shape used for external tanks and other bodies of revolution such as spheres or cylinders with dome-shaped ends. These tanks also are usually made of a weldable aluminum alloy. In this study, a detailed study is made of the irregularly shaped tank using an existing aircraft tank for comparison, and a more general survey is made of the possible applications of reinforced plastics to the symmetrical tanks.

The fuel tank used in the L-19 aircraft is considered to be typical of the irregularly shaped tanks used in a large number of Army fixed- and rotary-wing aircraft. It is fabricated of .040 thick 5052-0 aluminum in two halves which are joined by welding. Bosses for vent lines, a filler neck, gage mounting, etc., are welded to the tank. One baffle is welded inside the tank.

Several reinforced plastic materials are feasible for use in fuel tanks. Fiberglass reinforced epoxy or polyester resin laminates are considered to be most appropriate. Both have good wet and dry strength characteristics and fatigue properties and exhibit excellent compatibility with hydrocarbon fuel and oils. A discussion in another section of this report is presented on fuel compatibility. Epoxy laminates have the best properties but are more expensive than polyester and have more associated fabrication problems. The additional cost is warranted for this application; however, completely satisfactory tanks can be made with polyester resin. The fuel cells used on the F8U-2NE aircraft are fabricated with glass cloth preimpregnated with polyester resin.

A plastic laminate thickness of .040, which is the same as the aluminum thickness, can be used for this tank. The critical design loading on tanks such as that being considered here is normally the hydrostatic pressure loading on the flat portions of the tank. Since the geometry of the plastic tank is the same as the original tank geometry, the hydrostatic

pressure loads will be the same. A comparison of the mechanical properties of 5052-0 aluminum and the 181 epoxy laminate shows the ultimate strengths of the two materials to be roughly the same, but the aluminum has very low yield strength. The plastic laminate has a modulus of elasticity approximately one-third that of the aluminum. The use of a plastic laminate the same thickness as the aluminum (.040) will provide a structure working at a relatively low stress level; however, the deflections will be approximately three times as great for identical geometry. The deflections have not been evaluated numerically but are undesirable or unacceptable if they interfere with the function of the aircraft. The deflections can be reduced by the use of deeper stiffening beads on the flat areas or by use of additional stiffeners.

Vibration and vibratory stresses must be investigated in the detailed design of the tank. As indicated in the section of this report on dynamics, glass reinforced plastic materials have a greater degree of damping than do metal structures. This factor, coupled with careful consideration of natural frequencies and exciting frequencies during design, should eliminate problems due to vibration.

A concept of a reinforced plastic design of the L-19 fuel tank is shown in Figure 41. The basic tank is similar to the existing metal tank. It is made in two halves bonded together. The illustration shows four plies of 181 fabric impregnated with epoxy resin. In this particular tank, the two halves could be made on the same male mold. A wet lay-up with curing pressure applied by vacuum bag or autoclave is recommended for low quantity production. Matched metal dies could be used if the quantity is sufficient to justify the cost.

Preimpregnated fabrics or "Spiralloy Mat" manufactured by Hercules Powder Company can be used also. To date, epoxy preimpregnated materials are not adaptable to vacuum-bag molding but can be used when higher pressure can be applied. "Spiralloy Mat" is filament wound in a cylinder to any desired thickness, "B" staged, removed from the mandrel, and used in the same manner as other preimpregnated fabrics. Its main advantages are high strength properties and excellent drapability.

The fabrication technique will require some development work to optimize the method. Fittings could be molded in the laminate or attached by bonding or mechanical attachments after the tank shell is molded. Baffles inside the tank are bonded in place. The two half shells are bonded together around the periphery of the tank. Additional mechanical fasteners can be used at wide spacing in the joint to insure joint reliability.

The calculated weight of the fuel tank is approximately 6.5 pounds as compared to 10 pounds for the existing aluminum tank. This weight comparison is based on the tank wall surfaces and does not include the bosses and fittings which would be about the same for both tanks.

Tanks having shapes which can be generated as figures of revolution are adaptable to the filament winding process for fabrication. Filament-wound

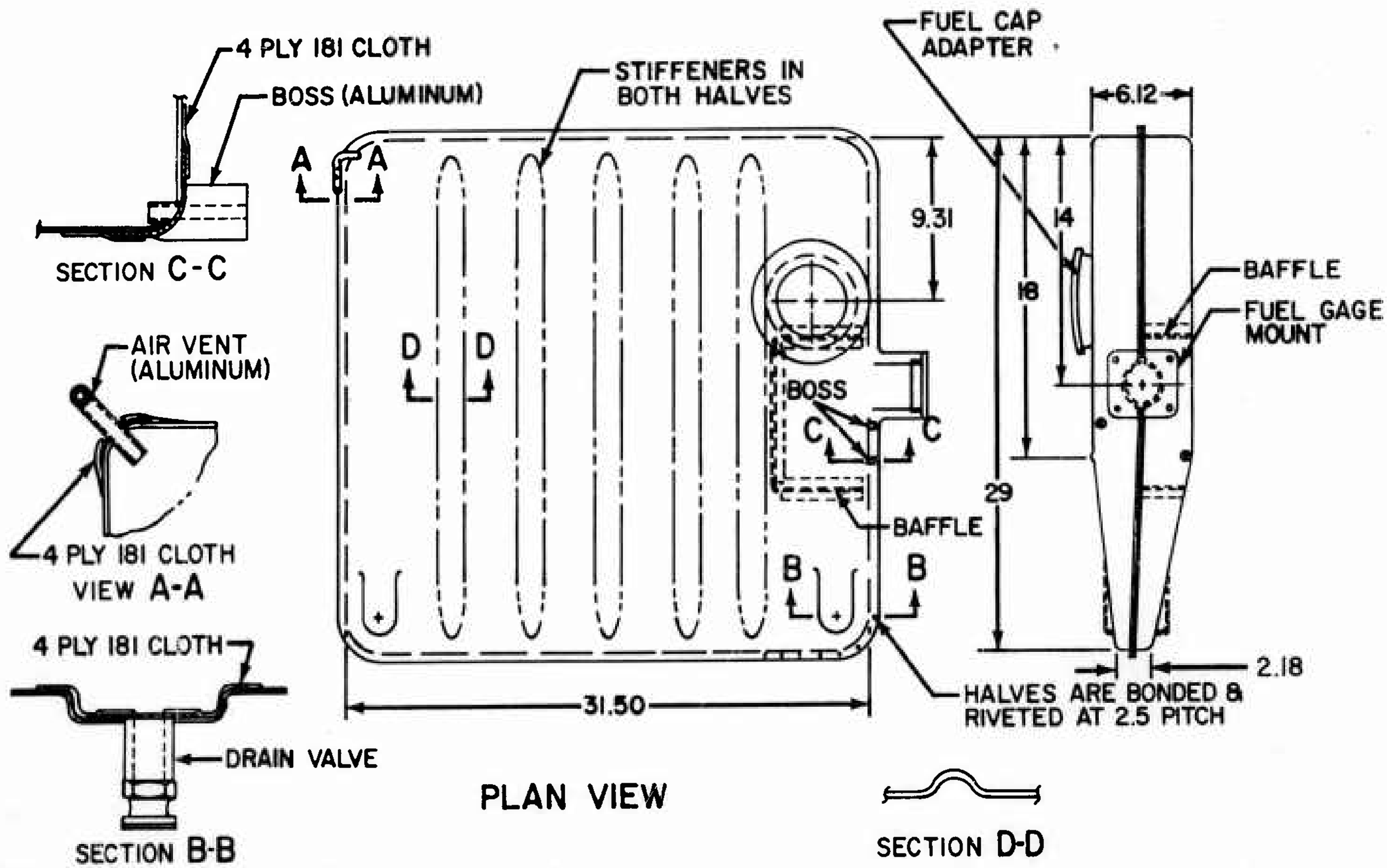


FIGURE 41. L-19 FUEL TANK OF REINFORCED PLASTIC

containers perform to best advantage when acting as pure pressure vessels where the stresses are primarily tension. Fuel and oil tanks are not loaded as pure pressure vessels, although hydrostatic pressures build up in portions of the tank during maneuvering of the aircraft. It is doubtful that full utilization of the high strength of the filament-wound structure can be realized because of the practical limitations on the minimum thickness of the material (handling, deflections, etc.). Wall thicknesses of less than 0.03 inch do not appear to be practical, and a thickness of 0.04 inch is more common. Wall thicknesses of aluminum tanks are usually in the range of 0.04 inch to 0.08 inch. Thus, weight savings in most cases will be due primarily to the lower density of the plastic material rather than to reduction in wall thickness. The plastic laminate weighs approximately 30 percent less than aluminum.

EVALUATION

Fuel tanks and other liquid containers for Army fixed-wing aircraft and helicopters can be fabricated from reinforced plastics, either laminated or filament wound, depending on the tank shape. Laminating should be employed in irregularly shaped tanks using woven glass fabric and epoxy or polyester resins. Filament winding would be applicable to regularly shaped tanks such as wing-tip or pylon tanks which have full figure of revolution configurations. The advantages and disadvantages of reinforced plastics versus aluminum tanks are outlined below.

Advantages

1. Superior corrosion resistance.
2. Potentially less fatigue stresses.
3. Lighter weight.
4. More easily cleaned.
5. More adaptable to irregular contours.

Disadvantages

1. Attachment of fittings appears to be more difficult and would require some development.

2. Greater deflections for given load and material thickness.

3. Tooling cost for filament winding is high.

The estimated cost of the reinforced plastic tank for the L-19 is $470.00 each if 10 are built, reducing to $240.00 each for a quantity of 100. No comparable information on the cost of the aluminum tank is available.

It is concluded that the use of glass reinforced plastics for fuel tanks and other liquid containers is feasible. Plastic fuel tanks will be less susceptible to corrosion damage and fatigue damage. Some specific designs with irregular contours may be fabricated more economically in plastic laminates than in metal. Weight saving potentials are minimized by the

lower modulus of elasticity of the plastic, necessitating designs having excessive strength in order to provide adequate rigidity in some configurations.

It is recommended that reinforced plastics be considered for present and future aircraft and helicopter fuel tanks. Each installation should be evaluated individually for the advantages and disadvantages of reinforced plastics in the particular application.

ROTOR AND PROPELLER BLADE DESIGN STUDY

The effort that could be allocated from this study program was insignificant compared to the total effort that has already been expended by others to investigate this highly specialized application of reinforced plastics. Therefore, this study summarizes the very meager available information. Considerable development work is known to have been accomplished by Vertol, Kaman, and Parsons on helicopter rotor blades and by Curtiss-Wright on propeller blades. Detail data on recent work in this field have not been made available to Hayes for this study.

In 1955 Vertol Division, or The Boeing Co. (Then Diasecki Helicopter Corp.) conducted a development program for the U. S. Army and Air Force in which fiberglass rotor blades for the H-21 were designed and tested (Reference 60). It was concluded that glass reinforced plastics are feasible for this application and that higher margins of safety for equal weight, or lighter weight for equal life, could be obtained for less cost than metal or wood blades. Blades are presently being tested for the HC-1B helicopter.

The Kaman Company has produced and tested a satisfactory fiberglass rotor blade for their H-43B helicopter. It is reported that it shows a 70 percent higher endurance limit than the standard blade at a cost substantially less. This blade was developed by substituting reinforced plastic for wood and metal components progressively. As each step was proved by test, other components were replaced with fiberglass and thoroughly tested. Finally, a successful complete fiberglass blade resulted. The present blade that is in production for the HU-2K helicopter uses an aluminum alloy spar with all-fiberglass trailing edge and skin wrap. Commitments from the procuring agency are being prepared to "phase in" a fiberglass blade for a follow-on program for the H-43B helicopter.

The nose "D" section of the all-fiberglass blade is a solid laminate of selectively oriented "Scotchply". The trailing edges of both types of blades are thin skins stiffened by full-depth ribs. The premolded skins and ribs are bonded together, and this subassembly is bonded to the spar.

Kaman has accomplished a considerable amount of strength and fatigue testing in developing these blades. All components have been subjected to repeated loads simulating actual operating conditions. The fiberglass blades indicate longer life than the conventional metal and wooden blades. The work of Kaman indicates rather conclusively that reinforced plastics are feasible materials for helicopter rotor blades.

No data are available on the work being conducted by the Parsons Aircraft Company.

A fiberglass propeller has been developed and tested by Curtiss-Wright Corporation's Propeller Division for V/STOL aircraft. It is reported that this blade has proved to be highly successful, weighs about half of equivalent hollow steel or aluminum alloy blades, and is low in cost (Reference 8).

IMBEDDED CONDUCTORS

Fabrication methods employed with reinforced plastics are such that electric wires or tubing for liquids and gases may be imbedded in the laminates or core of the primary structure during manufacture. This unique feature of reinforced plastics makes it possible to produce structural components with much of the plumbing and wiring "built-in" as an integral part of the structure. This technique has some desirable features, but a variety of technical problems require evaluation before adoption of the technique as a production procedure. Some applications of built-in wiring known to be in existence today are an imbedded antenna in a reinforced plastic tow target built by Hayes International Corporation and imbedded heating elements in reinforced plastic missile containers.

Various techniques may be employed in imbedding wires or tubing within the reinforced plastic. They may be introduced in the lay-up operation, or, in the case of sandwich sections, holes may be provided in the core for introuduction of the wire or tube before or after the core is combined with the face plies. In the latter case, some means must be found to seal completely the opening in which the wire or tube is installed to prevent the entrance of moisture. The following diagrams illustrate possible designs for inclusion of wires and tubes in reinforced plastic.

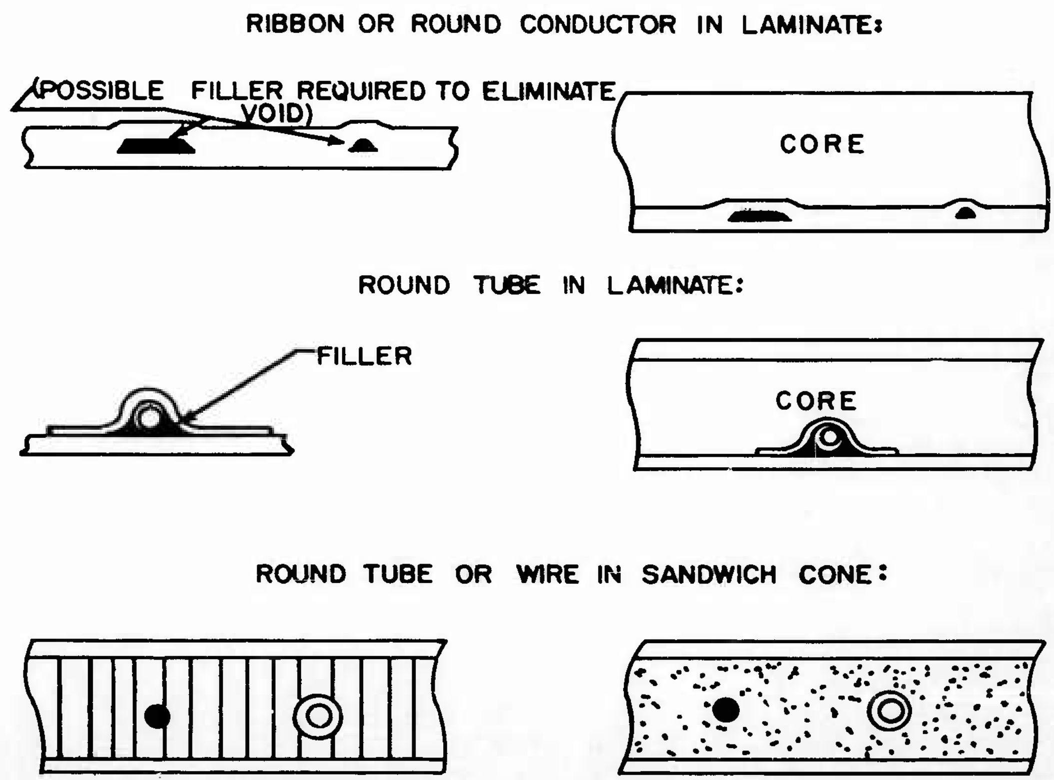


FIGURE 42. IMBEDDED WIRES AND TUBES IN REINFORCED PLASTICS

The primary advantage seen for imbedded wiring and tubing is that these items of hardware become an integral part of the structure, hermetically sealed from moisture, corrosion and fungus, and protected from normal wear, abrasion and scuffing. This will eliminate the need (and the possibility) of frequent inspection of this hardware and will reduce manhours required for inspection and for replacement of defective parts. However, the inaccessibility inherent in this design makes it doubly important that all phases of this technique be investigated to insure satisfactory service from the part after installation.

The effect of the difference in the thermal expansion properties of metals and reinforced plastics may be a source of difficulty in the use of imbedded wires and tubes in aircraft structural components. When two materials having different coefficients of thermal expansion are bonded together and then subjected to a change of temperature, one material tries to contract or expand more than the other, thereby inducing loads and stresses in both materials.

Table 14 lists the coefficients of thermal expansion for reinforced plastics utilizing various resins and the coefficients for metals commonly used in aircraft wiring and tubing. As an example of the effects of the difference in coefficients of two materials bonded together, assume that a copper wire has been imbedded between the plies during lay-up of a 20-foot-long wire panel. The difference in coefficients of expansion between these two materials is $(10.0 - 5.0)\ 10^{-6} = 5.0 \times 10^{-6}$ if an average value of 5.0×10^{-6} is assumed for the plastic. When this assembly is put in an oven at 370°F for curing, the copper will expand (5.0×10^{-6}) (20) (12) (300°) = 0.36 inch more than the reinforced plastic. Upon cooling, the copper will shrink more than the plastic, setting up loads and stress in both materials.

The stresses developed by thermal conditions may be approximated using known relationships as follows. The metal, upon cooling, will contract, inducing tension in the metal because of its restraint by the plastic. This, in turn, induces compression in the plastic surrounding the metal.

For equilibrium, the load induced in the metal must equal the load induced in the plastic, and the change in length must be the same in each material. Equating the elongation in the plastic to the elongation in the metal,

$$\alpha_p \ \Delta t \ 1 + \frac{P \ \ell}{A_p \ \ E_p} = \alpha_c \Delta \ t \ 1 \quad \frac{P \ \ell}{A_c \ \ E_c}$$

where

α	is thermal coefficient of linear expan. (In./In./F^{o})	
t	is temperature (F^{o})	P is load (lb.)
ℓ	is length (in.)	A is area (sq. in.)
E	is Young's modulus (lb/sq. in.)	

solving for load,

$$P = \frac{(\Delta t)\ (\alpha_c - \alpha_p)}{\dfrac{1}{A_p \quad E_p} + \dfrac{1}{A_c \quad E_c}}$$

Table 15 gives the load P and the corresponding stresses computed using this formula. Note that stresses are computed with varying areas of plastic corresponding to each wire size checked. This is done to show the effect of changing the assumption of effective plastic area.

The stresses computed for the copper are generally above the yield stress for annealed copper, indicating that some slight plastic deformation may take place. Also, the shear stress shown for the laminate (assumed 1/32 thick) attaching the tube is beyond the 1400-1500 p.s.i. range listed (Reference 38 as allowables for typical materials which might be used here. The use of a slightly thicker laminate attaching the tube would reduce the shear stresses to an acceptable level.

Another phase of thermal reactions of two different materials subjected to temperature change is the possibility of bowing of the structure caused by the different rate of expansion of the metal and the laminate. No specific analysis of this effect is attempted; however, it should be considered in designs having a relatively large mass of metal attached eccentrically to the center of area of the plastic.

A major problem associated with the use of imbedded wiring or tubing in structural components subjected to flight and landing loads is the effect of the difference in Young's modulus of the two materials. Young's modulus by definition is the ratio of stress to strain in the elastic range of the material. Young's modulus and the elongation for some typical aircraft construction materials are given in Table 13.

TABLE 13
COMPARISON OF METAL PROPERTIES

Metal	Young's Modulus E $\frac{\text{lb.}}{\text{sq. in.}}$	Yield Stress f_y $\frac{\text{lb.}}{\text{sq. in.}}$	Elongation %
5052-0 Aluminum	10.5×10^6	10000	16-20
2024-T4 Aluminum	10.5×10^6	46000	10-12
Type 316 Corrosion-Resistant Steel	28.0×10^6	30000	30-40
Annealed Copper	17.0×10^6	4-6000	35-40

TABLE 14

COEFFICIENT OF THERMAL EXPANSION

Material	Temp. °F	Coefficient of Thermal Expansion - In./In./°F x 10^{-6} Reinforced Plastic Parallel To Warp	Perpen. To Warp	45° To Warp	Metals
Polyester Resin	-100 to 100	7.8	9.3	8.5	-
(MIL-R-7575)	200 to 400	1.4	2.3	1.3	-
Epoxy Resin	-100 to 200	5.5	6.7	6.7	-
(MIL-R-9300)	300 to 600	3.3	1.5	2.3	-
Aluminum Alloy	-58 to 68	-	-	-	12.1
	68 to 212	-	-	-	13.0
	68 to 392	-	-	-	13.5
Corrosion	0 - 200	-	-	-	8.7
Res. Steel	0 - 600	-	-	-	9.3
(302, 316)					
C. R. Steel	0 - 200	-	-	-	5.7
Type 416	0 - 600	-	-	-	6.1
Pure Copper	32 - 572	-	-	-	10.5

TABLE 15

THERMAL STRESSES

Wire Diameter	Plastic Diameter	Load P (Lb.)	Tension Stress in Metal (p.s.i.)	Comp. Stress in Plastic (p.s.i.)	Shear Stress in Bond (p.s.i.)
1/16	3/16	46	15000	1850	233
1/16	1/4	57	18400	1230	290
1/8	3/16	59	4800	3840	150
1/8	1/4	108	8850	3000	280
1/8	3/8	176	14300	1800	450
3/16	7/16	312	11300	2540	530
3/16	1/2	366	13300	2170	620
3/8 O.D. Tube, 1/32 Wall	7/16	149	4420	3740	Shear Stress In Laminate 2390

Since the modulus of all the metals is higher than the modulus for the plastic (3.0×10^6), the metals will load-up faster than the plastic up to the yield point of the metal. The action of the metal then become inelastic.

In the case of a small copper wire in a laminate, the area of the wire will be small proportionate to the plastic and, because of its low yield strength, will have very little effect on the load and elongation in the plastic. The actual elongation that must take place in the metal can be estimated by computing the strain in the reinforced plastic. If a stress of 50,000 p.s.i. is assumed in the plastic, the strain is $50000/3.0 \times 10^6 = .016$ in/in. This indicates that the inelastic elongation in the metal will be slightly less than 1.6 percent (a small amount of elongation takes place before the yield point of the metal is reached). Table 13 shows that the potential elongation of the copper ranges from 35 percent to 40 percent, indicating that failure of the metal will not take place under nonrepeating loads. However, work-hardening of the copper will occur under repeated load conditions and could eventually cause a break in the wire.

A 2024 T-4 aluminum tube bonded in a plastic laminate under load would, because of its high yield strength, materially affect the load distribution between the metal and the plastic. The aluminum would, however, load-up to its yield point while the stress in the plastic was relatively low. The stress in the plastic corresponding to the yield stress in the aluminum would be $46000 \left(\frac{E \text{ plastic}}{E \text{ alum.}} \right) = 13000$ psi.

It thus becomes obvious that if the stress level in the aluminum is held below the yield point, the plastic will be used inefficiently. If the plastic is loaded to its capacity, the aluminum will be loaded beyond its yield strength and will undergo area reduction with possible loss of bond. It may also work-harden and fail under repeated load conditions.

The discussion up to this point has been based on having a good bond between the plastic and metal such that the plastic and insert deflect as one piece. The load transfer due to the bond between the two materials appears to have undesirable effects which could be eliminated by preventing a bond between the two. With proper design of end fittings on the inserts to allow relative movement, the structure could be designed as all plastic with the inserts treated as holes in the plastic. It is believed that this approach is more practical than trying to bond the insert to the plastic.

EVALUATION

The concept of imbedding the wiring and tubing of a reinforced plastic structural component within the structure itself has definite merit from the standpoint of improved serviceability and reduced maintenance cost. Certain problems caused by differing thermal expansion and differing relative stiffness for reinforced plastics and metals require experimental investigation by means of test specimens.

Advantages derived from this method of construction may be summarized as follows:

1. Elimination of damage to wiring and tubing caused by vibration and abrasion, by movement of personnel, or by careless handling of cargo.

2. Better protection from moisture, corrosion, and fungus.

3. Reduced maintenance time realized by eliminating periodic inspection of wiring and tubing.

4. A cleaner, more accessible cockpit and cabin area.

5. Less time required for replacement of large structural components in the field. (This is based on the assumption that the conventional procedure is to replace the structure only, with the wiring and tubing being transferred from the old assembly to the new. Obviously, if both assemblies are complete in themselves, the only advantage is in the better protection of the wiring and tubing from damage due to handling.)

6. Possible reduction in weight through use of bare wires with the reinforced plastic acting as insulation.

Disadvantages and questionable areas resulting from this method of construction are:

1. Bonding of the commonly used metals to structural components subjected to high stresses will in most cases result in plastic working of the metal, which may result in failure of the bond or the metal.

2. Temperature changes in metal-plastic combination structures may result in unacceptable bowing of the structure.

3. Faulty or damaged wiring and tubing will be more difficult to repair.

4. Routing of wiring and tubing will require much more detailed planning prior to installation, since the routing cannot be changed after fabrication.

5. Changes in equipment in aircraft in service will be somewhat hampered by the fixed nature of the wiring and tubing.

6. More connectors of more complex design may be required.

7. Fabrication of components with imbedments will increase fabrication cost. However, cost of stringing wires and tubing in aircraft assemblies will be minimized, which will, at least partially, offset additional fabrication costs.

CONCLUSIONS

The design study leads to the following conclusions:

1. The concept of imbedding the wiring and tubing of a reinforced plastic structural component within the structure itself has definite merit from the standpoint of improved serviceability and reduced maintenance costs and appears to be feasible.

2. A test program is essential for further evaluation of this concept. If the program reveals no adverse effects due to stress in the plastic or thermal properties, additional effort should be expended in investigating repair problems, optimum connector design, etc.

3. Further investigation should be made of methods of preventing a bond between the imbedment and the plastic and the design problems involved. This may require coating the imbedment with some substance which will prevent adhesion but will not contaminate the resin or reduce the bond strength in the laminate.

RAIN EROSION

Rain erosion is a type of surface damage encountered along the leading edges of exterior reinforced plastic aircraft parts such as wings, radomes, nose sections and other similar components. It is caused by the impact of water droplets upon the surfaces when flying through rain. The problem becomes more serious as the speed increases. Water droplets impinging at a high velocity against a reinforced plastic surface produce cavities in the material. This produces a radially outward flow at a velocity considerably greater than the original impact velocity. The resulting repeated stresses in the cavity continue the erosion process. At low speeds, a large number of droplet impacts is required to produce erosion; however, as speed increases, the number of impacts required decreases.

The rate of rain erosion is a function of velocity, droplet size, the number of impacts, material, and surface conditions. Tests at Cornell Aeronautical Laboratory (Reference 59) have indicated that the time of exposure required to produce a given amount of erosion is inversely proportional to the eighth power of the velocity in the speed range of 250 miles per hour to 600 miles per hour. Laboratory tests also show that an average drop size of 2.5 millimeters produces erosion three or four times faster than does an average drop size of 1.9 millimeters. The effects are greatest on surfaces normal to the line of flight, becoming less severe on surfaces forming angles smaller than 60 degrees with the line of flight and appreciably reduced or negligible at angles of 15 degrees or less.

A considerable amount of research has been accomplished on the problem of rain erosion, primarily by the Wright Air Development Center. The following table summarizes the rain erosion properties of plastic materials and coatings as reported in Reference 59.

TABLE 16
RAIN EROSION PROPERTIES OF MATERIALS EVALUATED
AT 500 MPH THRU 1"/HOUR SIMULATED RAINFALL

General Material Description	Time To Initiate Pitting	Time To Erode Through
1/8 Inch Epoxy Laminate	2-3 Min.	30-50 Min.
1/8 Inch Polyester Laminate	30 Sec.	5 Min.
1/8 Inch Phenolic Laminate	15 Sec.	3 Min.
* 10 MIL Neoprene Coating	-	50-70 Min.
* 10 MIL Silicone Coating	-	10 Min.
* 10 MIL Polyurethane Coating	-	50 Min.
* 10 MIL Gates White Neoprene	-	50-70 Min.
**Alumina		300-420 Min.

* Coating over a typical polyester laminate.
**Moderate pitting.

Unpublished data from The Martin Company showed that a void free glass fabric-polyester laminate mounted on the leading edge of an airplane exhibited the following erosion damage after flight through rain at 475 miles per hour:

Exposure Time	Extent of Erosion
5 Minutes	Slight pitting of the resin.
10 Minutes	Eroded through surface resin to first ply.
15 Minutes	Erosion through one or more plies.

At a speed of MACH 2, a comparative laminate eroded through three to four plies in one-half second exposure time.

Other sources of information indicate similar effects of rain erosion. It can be concluded that leading edge surfaces of components of Army aircraft operating in the higher speed ranges should have some protection to reduce rain erosion. Epoxy resin laminates have considerably higher erosion resistance than polyester laminates, but the thin laminates that are otherwise practical to fulfill structural requirements would probably be damaged more severely than thicker laminates. These surfaces should have sufficient strength to give a good backing for the protective coating.

At subsonic speeds, reinforced plastic surfaces can be adequately protected by coating with a 10 - 20 mil layer of neoprene. Neoprene coatings will give several hours of protection during flights through rain at 500 miles per hour. Extremes of temperature affect neoprene coatings adversely. Temperatures in excess of 200 degrees Fahrenheit cause brittleness and reduce adhesion in the coatings. At supersonic speeds, neoprene coatings behave as hard, brittle materials and are no longer erosion resistant.

Considering the speed range of Army aircraft, a neoprene coating will probably provide adequate protection for affected areas of all surfaces with the exception of helicopter rotor blades and propeller blades. Other coatings such as the polyurethane type should be further investigated. The leading edges of these components will undoubtedly require stainless steel cover plates for protection against damage from miscellaneous solid particles as well as rain erosion.

HYDROCARBON FLUID COMPATIBILITY

The response of reinforced plastics to various environmental conditions is dependent upon both the nature of the environment and the composition of the resin and reinforcement used. The compatibility of low-pressure reinforced plastics to a hydrocarbon fluid environment is influenced by the type of fluid and by the duration and temperature of exposure.

In general, epoxies exhibit the best resistance to most organic compounds used in aircraft. Polyesters are generally next best, with a polyester-epoxy hybrid providing excellent fuel resistance. Phenolics and silicones have only moderate resistance to a variety of organic fuels and lubricants.

Those organic liquids most closely associated with aircraft are grade 115/145 aviation gasoline; JP-4 and other jet fuels; and lubricating and hydraulic oils. In considering the fabrication of aircraft fuel tanks, crankcases and hydraulic oil chambers from reinforced plastics, the designer must be concerned with both the effect of the organic liquid on the plastic and vice versa.

Some investigation has been made into the effect of long-term exposure of fuels and oils to various low-pressure plastic laminates. The Bureau of Ships, Department of the Navy sponsored a study (Reference 9) in which the effects of four shipboard fuels upon nine polyester and five epoxy resin laminates were reviewed. Sample panels of the various laminates were tested after being immersed in the four fuels for a period of six months at 110 degrees Fahrenheit. The four fuels were grade 115/145 aviation gasoline, JP-4 jet engine fuel, diesel fuel, and Navy special fuel oil. The following general conclusions were drawn as a result of these immersion tests:

1. The gum content of aviation fuel was increased an average of 2.5 mg/100 ml by the polyester materials but was unaltered in the presence of the epoxy materials. Since the gum content of aviation fuel increases with age, this amount is not considered to be harmful.

2. The JP-4 jet engine fuel, diesel and Navy special fuel oil were unaltered after six months by the presence of the fourteen types of laminates.

3. Immersion in the fuels did not affect the hardness of the plastic materials.

4. The epoxies were least affected by the fuels and did not significantly affect the quality of the fuels.

5. Resins containing fire retardant compounds exhibited more uniform results and had less adverse effect upon the fuels than did the resins without the fire retardant.

WADC Technical Report 57-674 (Reference 64) presents data on the "effects on the mechanical and physical properties of five typical government specification plastic materials after immersion in four experimental high-temperature hydraulic fluids, under various temperature and time of exposure conditions". Polyester, heat-resistant polyester and phenolic laminates were immersed in four hydraulic oils for 24 hours and 7 days at room temperature and for 24 hours at 160°F, 250°F, 400°F and 500°F. As a result of these tests, the following general conclusions were drawn:

1. The immersion tests had negligible effect upon the mechanical strength properties of the three types of plastic laminates.

2. All three types of laminates were checked for thickness change in accordance with the applicable Government specifications, with the following results:

   a. Polyesters showed no significant change.

   b. Heat-resistant polyester exceeded the maximum increase in thickness of MIL-R-25042.

   c. The thickness change exhibited by phenolic laminates when immersed at room temperature and at 250°F was within the maximum allowed by MIL-R-9299. However, tests conducted at 400°F and 500°F exceeded the maximum allowable by the specification.

3. The three types of laminates tested for weight change were below the maximum allowed by the applicable Government specification.

It is generally concluded that the hydrocarbon fluid resistance of reinforced plastic laminates falls in the following order of decreasing resistance; namely, epoxies, polyesters and phenolics. It may also be generally concluded that, in the presence of most hydrocarbon fluids, epoxy, polyester and phenolic laminates:

1. Are either unaltered or are only slightly reduced in strength;

2. Are unaffected in hardness;

3. Increase in thickness slightly;

4. Increase in weight slightly, depending greatly upon the fluid and the plastic involved.

The available data indicate that aircraft fuels and oils do not have a serious detrimental effect on reinforced epoxies, polyesters, and phenolics; nor do these resins affect the fuels and oils to a degree that would be harmful. Additional tests for evaluation of specific applications are desirable. Tests to determine fatigue characteristics with long-term exposure to hot lubricants are especially recommended.

Glass cloth reinforced polyester laminated tanks have been used for many years to contain lubricating oils on aircraft. The B-50 is one example of an aircraft with long service life having reinforced polyester oil tanks.

The necessity of resisting special chemicals needs further study to establish definite data for use by design engineers. One approach which should assist reinforced plastics in special chemical resistance is the use of overlay plies of material like neoprene or dynel in the laminated part. It is known that materials of this type may be cured with the glass reinforced plastic and become an integral part of it. Further study is required in this area.

ACOUSTICS

Extensive surveys have been conducted, Reference 71, to measure the actual sound levels present in Army aircraft. These surveys show that very high sound levels do exist on all Army aircraft. The sound levels were measured at many locations in and around the aircraft. The highest noise levels exist on helicopters and originate primarily from the main rotor, engine, and transmission. Noise levels of 100 decibels are found at many locations, and some noise levels have been measured at peaks up to 120 decibels.

High sound levels in and around Army aircraft are detrimental to the successful use of the aircraft for the following reasons:

1. Ease of detection by the enemy.
2. Induced vibration damage to instruments and structure.
3. Interference with communication of aircraft crew.
4. Harmful effect on ground personnel.
5. Potential hearing impairment of operating personnel.
6. Probable reduced efficiency of operating personnel.

The use of fiberglass reinforced plastics for the construction of primary structure in Army aircraft raises the question of the ability of fiberglass reinforced plastics as a construction material to reduce the relatively high noise levels present on Army aircraft. First reaction to this consideration is that fiberglass reinforced plastics should certainly help to reduce noise levels because of its damping characteristics and the wide use of glass fibers for sound and thermal insulation. Further inquiry will lead one to search technical literature to find some test data to verify the intuitive feeling that fiberglass reinforced plastics structure should inherently attenuate noise in Army aircraft. At this point, the literature search reveals that essentially all acoustical research work on construction materials has been done to evaluate their use as architectural and insulation materials and that high-strength fiberglass reinforced plastic materials have not been examined for their acoustical properties.

Noise levels can be reduced either by reducing the source of the noise or by absorption of the noise using "acoustically designed" construction materials. The reduction of the noise source is a design consideration which is beyond the scope of this study program. The contribution which reinforced plastics as a structural matèrial may contribute to the noise reduction in Army aircraft led to a review of the basic principles which govern the acoustical properties of a material. Sound may be transmitted from its source to the observation point by one or more of the following paths:

1. Direct air transmission
2. Transmission through construction material
3. Source induced vibration in surrounding construction material
4. Induced vibration through source mounting

Direct air transmission (item 1) is reduced in reinforced plastics due to the relatively large single-piece-type construction which helps to eliminate the air gaps usually present with the numerous mechanical attachments characteristic of metal airframe construction. The vibration through the source mounting (item 4) is a mechanical design problem involving optimum shock mounting configurations. Items 2 and 3 are primarily governed by the inherent acoustical characteristics of the construction material. Assuming that the structural material is a solid, the inherent material properties which influence the acoustical characteristics of a structural part are:

1. Stiffness
2. Mass
3. Internal damping
4. Surface absorption

The acoustical characteristics of a reinforced plastic structural part are dependent on the interaction of the four factors noted above as well as on the design details of geometry, attachments, etc. Stiffness is dependent on the modulus of elasticity of the material. The effect of mass has been defined in a "weight law". The "weight law" states that the average sound transmission loss for a homogeneous partition depends on its mass per square foot, the heavier the better, in logarithmic proportion. Internal damping is dependent on the ability of a material to dissipate mechanical vibration energy in the form of heat rather than to radiate this energy as airborne noise. The surface absorption of a structural material is directly related to the degree and type of surface porosity present.

The most significant property of reinforced plastics which contributes to noise reduction is internal damping. The attribute of internal damping is virtually nonexistent in most metals. This property can be measured comparatively by determining rates of decay of vibrating specimens and the sharpness of the resonance curve, and by other methods. To date, no test data have been found which quantitatively evaluate the internal damping of highstrength reinforced plastics. As much as these data would be desirable from an academic standpoint, their absence is not vital because any useful acoustical analysis has to deal with all of the material and design variables which influence the acoustical characteristics of a particular part.

There were plans to conduct sound transmission tests on reinforced plastic specimens. The objective was to measure the sound transmission characteristics of several varieties of reinforced plastic specimens and to compare them to each other and to metal. It was thought that accurate data could be obtained by measuring sound transmission loss on test panels. A search for acoustical testing facilities in the eastern United States revealed that the only facilities commercially available were at the Martin Co. in Baltimore, Md., and the O-C Fiberglas Corp. in Columbus, Ohio. A visit was made to the Martin Co. and to the Research Laboratory of the Owens

Corning Fiberglas Corp. Examination of the available test facilities and discussions with the acoustical test engineers at Martin and Owens Corning showed that the facilities could measure only sound transmission loss in flat panels mounted in a wall between a sound chamber and an anechoic chamber. Analysis of this testing method shows that only comparable data can be obtained, since the panel size, mounting method, and test chamber characteristics are all variables which prevent absolute measurements of sound transmission loss. With the threatened lack of useful reference data on the transmission loss characteristics of reinforced plastics, it was decided to delete that type of testing from this program. It became evident that it is necessary to obtain two types of acoustical data which would be useful for further study of the acoustical characteristics of aircraft structures.

One is vibration analysis of test strips of reinforced plastics and other materials to measure the decay rate of the material in units of absolute measurement which could be used in acoustical analysis work. These tests would be made on suspended test panels which would produce basic data that were not influenced by variables other than the inherent material characteristics.

The other type of acoustical testing needed is the analysis of the sources and paths of sound in Army aircraft. Although many measurements of sound levels have been made on Army aircraft, these measurements have not defined such important factors as the amount of noise at a certain point which is air-borne and that which is structure-borne, the effect of induced sound vibrations on the structure as compared to direct excitation from mechanical sources, and the improvement factors to be realized by selective isolation of structural areas to reduce acoustical noise. It is vital that these considerations be studied to produce a more complete understanding of the proper approach needed to reduce acoustical noise in Army aircraft. A direct approach to the continuance of the study to reduce noise levels in Army aircraft would be to construct several major structural pieces of a particular aircraft from reinforced plastic and then to measure noise level reductions which resulted from this modification. This approach would be needed to prove the predictions of an alalytical study of noise reductions. Therefore, perhaps it would be more expeditious, economical, and even more accurate to proceed to the "proof of the pudding" first and cause the resulting data to be directly applicable to an operating aircraft.

Intuitively, the information at hand indicates that the direct modification of an aircraft would not only be the most practical technical approach but also would produce more accurate and meaningful test data on the acoustical characteristics of reinforced plastics in Army aircraft.

Reinforced plastic structural components could make improvements in noise reduction on Army aircraft, but it is doubtful if these improvements would be significant. Their good internal damping characteristics have been

adequately displayed in practice if not by test data. Their ability to be made in relatively large single-piece construction will prevent direct air transmission of noise. The apparent weaknesses, acoustically, of mass and stiffness can be offset by judicious design which is greatly facilitated by the ease with which reinforced plastics can be made into sandwich structures and made with built-in stiffeners.

The determination of the degree of effectiveness of reinforced plastics in reducing noise was beyond the scope of this program. It is recommended that the study be continued, to include the following:

1. Conduct tests to obtain quantitative data on damping characteristics of reinforced plastics.
2. Determine the sources of noise and the transmission paths in selected aircraft.
3. Prepare a preliminary reinforced plastic design of the aircraft body structure and analytically evaluate noise transmission characteristics of both designs.
4. If it is concluded that the reinforced plastic design has possibilities of reducing the noise level, fabricate and test a full-scale component.

TEST PROGRAM AND RESULTS

The design studies accomplished in this program were based on analytical determination of strength and stiffness of the various reinforced plastic components. Many variables affect the performance characteristics of reinforced plastic structures. One of the greatest handicaps to a designer is the lack of adequate test data substantiating the analytical approach to structural design. The work that has been accomplished has usually been for a specific purpose and is proprietary. Therefore, there has not been sufficient coordinated test and experience data accumulated that can be used as accurate guidelines for design.

Of course the most conclusive proof of design is to fabricate full-scale components and test them under actual or simulated conditions and loads. This was not practical for the components investigated in this study. Therefore, the evaluation of the designs must depend on analysis and laboratory tests of specimens representative of detail problem areas.

A laboratory test program was conducted to substantiate specified design problems resulting from the design studies and to further the basic knowledge of reinforced plastics. The effects of various materials used in different types of construction were determined. These tests were in the following categories:

I. Compression Buckling of Panels

   A. Flat Sandwich
   B. Curved Laminates
   C. Curved Sandwich

II. Panel Shear

   A. Solid Laminate
   B. Sandwich

III. Bending of Flat Sandwich Panels

IV. Fasteners in Solid Laminates

   A. Bolted
   B. Riveted
   C. Bonded

V. Filament-Wound Tubes

VI. Serviceability of Thin-Faced Laminates

VII. Effects of Imbedded Conductors and Tubing in Laminates

Only materials that are most adaptable for use in fabrication of the components investigated were included in the tests. Materials for solid laminates and faces of sandwich panels included 181 type cloth and polyester resin, 181 type cloth and epoxy resin, and "Scotchply", a nonwoven fabric manufactured by Minnesota Mining and Manufacturing Company. Sandwich core materials included aluminum and fiberglass honeycomb, aluminum "Multiwave" and polyurethane foam.

Vacuum-bag molding was used on most of the specimens. "Scotchply" cannot be successfully molded with the low pressure obtainable from a vacuum bag. "Scotchply" laminates were molded in a press at the laminating pressure necessary to obtain the high strength characteristics of this material.

Test specimens for all attachment tests and for the determination of the effects of imbedments in laminates were fabricated at Hayes. The fabrication of all other specimens was subcontracted to Summit Industries, Gardena, California. All testing was accomplished in Hayes' Laboratory with the exception of bonded joint tests. Bonding of the joints using Hayes-furnished laminates and the testing of the joints were accomplished through the courtesy of Bloomingdale Rubber Company, Aberdeen, Maryland, and Minnesota Mining and Manufacturing Company, St. Paul, Minnesota.

A detail description of all specimens, the method of testing, and the results for each type test are included. Three each specimens were tested for each configuration of all tests. Some of the test results are also presented as curves of failing stress vs. number of plies. The points on the curves represent the average of the three specimens. All tests accomplished at Hayes were made using a Baldwin 3000,000-pound testing machine.

In the fabrication of the test specimens, a problem was encountered in getting good bonds between polyester faces and the sandwich cores. This problem is discussed in the section on fabrication of test specimens. Most of the test panels were completed before the test results were obtained and evaluated and too late to change the manufacturing procedure. The test results of the polyester-faced sandwich panels were, in general, lower than was expected. Summit Industries made additional test panels using material from another source to spot check the effects of the poor bonds. Test data for both materials are included. The points on the plots for panels of the alternate material is represented by an open circle, ○ , instead of the solid ● as for the other polyester.

Originally, the test program was to include an evaluation of tubes fabricated by various processes with several different materials. These tests were primarily for an evaluation of glass reinforced plastic for

power transmission shaft applications. Further study indicated that power transmission shafts are not feasible applications for reinforced plastic. Therefore, the significance of the tube tests decreased. Since tubes constructed by filament winding offer the greatest versatility in strength characteristics, it was decided to obtain data on the effects of some variables in this method of construction.

Filament winding of structural members is believed to offer potential advantages for some components. Data on thin-walled tubes will be quite beneficial as design data for miscellaneous filament-wound structures other than pressure vessels. See the section on filament winding for a further discussion.

The cost of procuring the required test specimens, fabrication of the test fixtures, and testing was found to be somewhat greater than the funds that could be allocated for these data. Hercules Powder Company, one of the pioneers in developing filament winding, has accomplished a considerable amount of testing similar to that proposed for this program. The desired data were purchased from Hercules. Included were data on bending, axial compression, shear, torsion and natural frequency of filament-wound, epoxy resin bonded fiberglass structures. All of these data were based on tests previously conducted by Hercules with the exception of the torsional data. Additional tests were conducted to obtain the torsional data for this program. All data obtained from Hercules Powder Company are summarized in this report.

FABRICATION OF TEST SPECIMENS

All test specimens used in this program with the exception of those used for attachment tests and imbedded conductors were fabricated by Summit Industries, Compton, California. Those for attachment and imbedded conductors tests were fabricated by Hayes. The methods and fabrication processes that were used are considered to be the same or represent methods of fabrication that would be used in the manufacture of actual aircraft components. In some cases, higher strength characteristics could probably be obtained if a different process was used. As an example, it is believed that higher strength sandwich could be obtained if the faces were precured under high pressure and then bonded to the core rather than the single-step vacuum-applied pressure method that was used. Vacuum-bag or autoclave molding is considered to be a feasible method of fabrication of aircraft components investigated in this program using sandwich construction.

The procedures and cure cycles used for the test specimens are, in general, standard practice and based on past experience. A summary of detail fabrication data is shown in Tables 17 and 18. No particular problems were encountered in fabrication except in the polyester-faced sandwich panels.

In order to eliminate variables in similar materials manufactured by different manufacturers, all materials of a particular type were to be from the same source. As an example, all type 181 polyester preimpregnated material was to be Moboloy 81D supplied by Cordo Chemical Company. Tests of sandwich panels using Moboloy 81D showed poor bonds between the faces and cores using FM-97 adhesive, Bloomingdale Rubber Company. It was found that sandwich panels fabricated in exactly the same manner but using a polyester preimpregnated fabric, #PGLA, manufactured by American Reinforced Plastic Company, had a flatwise bond tensile strength several times the strength of panels using Moboloy 81D.

A considerable amount of study, research and test was accomplished by Hayes, Summit Industries, Cordo Chemical Company, and Bloomingdale Rubber Company in an effort to determine why poor bonds resulted when Moboloy 81D was used with FM-97 adhesive and to determine if good bonds could be obtained by an optimum cure cycle. This was not done to prove that Moboloy 81D and FM-97 could be used. Neither should the poor bonds obtained the interpreted as an adverse criticism of the material. It is believed that this same problem could have resulted with material from other sources.

No experience could be found in industry where Moboloy 81D and FM-97 adhesive had been used to fabricate a sandwich component, nor did anyone have any reason to suspect that the combination would not give satisfactory results.

In the investigation, it was determined that the Moboloy 81D used was a low temperature curing resin. Panels were made with a high temperature curing Moboloy 81D and flatwise tension was somewhat improved, but only about one-half the value for American Reinforced Plastics preimpregnated material. Variation of the cure cycle affected the results, but an optimum was not developed.

It is believed that the low-strength bonds were due to a chemical action between the polyester in the Moboloy 81D material and the FM-97 adhesive that was different from that between the polyester in the American #PGLA preimpregnated and the FM-97. The American polyester is chlorinated and the Moboloy polyester is not. Bloomingdale has not experienced this problem before. They are investigating the chemistry and will recommend another adhesive if it is determined that the materials are not compatible.

Test data indicate that low-strength face to core bonds have a significant effect on the strength of the sandwich. Several additional panels of various types were fabricated and tested using American #PGLA preimpregnated fabric. In general, the new panels having a better bond exhibited higher strength. This was especially true of the flat compression buckling panel. In these panels the failing stress increased 100 percent. Curved compression panel strength also was much higher. The panel shear strength did not increase. The effect of the bond strength for panel shear is not conclusive.

TABLE 17
FABRICATION PROCESS - LAMINATED TEST SPECIMENS

Type Specimen	Spec. No.	Material	Process and Pressure	Cure Cycle
Panel Shear	239-1, 2, 3	181 Polyester Preimpregnated (1)	Vacuum Bag 12-14 p.s.i.	2 hrs. @ 300°F
	239-4, 5, 6	181 Epoxy Preimpregnated (2)	Vacuum Bag 12-14 p.s.i.	2 hrs. @ 200°F 1 hr. @ 250°F 1 hr. @ 300°F
	239-7, 8	Scotchply 1009 (3)	Vacuum Bag 12-14 p.s.i.	45 min. @ 330°F
Compression Buckling	242-1, 2, 3, 4	181 Polyester Preimpregnated (1)	Vacuum Bag 12-14 p.s.i.	2 hrs. @ 300°F
	242-5, 6, 7, 8	181 Polyester Preimpregnated (1)	Vacuum Bag 12-14 p.s.i.	2 hrs. @ 200°F 1 hr. @ 200°F 1 hr. @ 300°F
Laminates for Attachment Tests		181 Polyester Preimpregnated (1)	Press, 30 p.s.i.	Heat up to 300°F, 15 min. at 300°F
		181 Epoxy Preimpregnated	Press, 30 p.s.i.	10 min. @ 350°F with contact pressure, 15 min. @ 350°F @ 30 p.s.i.
		Scotchply 1002 (3)	Press, 30 p.s.i.	4 min. @ 330°F with contact pressure, 35 min. @ 330° @ 30 p.s.i.

(1) Type 181 cloth preimpregnated with polyester resin, Moboloy 81D, resin content 39%, Cordo Chemical Co.
(2) Type 181 cloth preimpregnated with Shell Chemical Co., 828 epoxy resin and RP7A catalyst, resin content 38-42%
(3) Scotchply - Minnesota Mining and Manufacturing Co.

TABLE 18
FABRICATION PROCESS FOR SANDWICH TEST SPECIMENS

Specimen Type	Specimen No.	Face Material Notes (2) (3) (4)	Core Material	Cure Cycle Note (1)
Panel Shear	240-1, 2, 3	181 Polyester Preimpregnated (5)	Aluminum Honeycomb	1 hr.@ 300°F, 1 hr.@ 330°F
	240-4, 6	181 Epoxy Preimpregnated	Aluminum Honeycomb	2 hrs. @ 200°F, 1 hr. @ 250°F, 1 hr. @ 300°F
	240-5	181 Epoxy Preimpregnated (5)	Aluminum Honeycomb	Faces precured 2.5 hrs @ 200°F sandwich 1 hr. @ 300°F, 1 hr. @ 330°F
	240-7	Scotchply 1009	Aluminum Honeycomb	Faces precured 45 min. @ 330°F in press @ 250 p.s.i.; sandwich 2 hrs. @ 330°F
	240-8, 9, 10	181 Polyester Preimpregnated (5)	Aluminum Multiwave	1 hr @ 300°F, 1 hr @ 330°F
	240-11, 12	181 Polyester Preimpregnated (5)	Aluminum Honeycomb	1 hr. @ 300°F, 1 hr. @ 330°F
	240-13, 14, 15	181 Polyester Preimpregnated (5)	Polyurethane Foam	4 hrs. @ 235°F
Compression Buckling Flat Panels	241-1, 2, 3, 4	181 Polyester Preimpregnated (5)	Aluminum Honeycomb	1 hr. @ 300°F, 1 hr. @ 330°F
	241-5	181 Polyester Preimpregnated (5)	Aluminum Honeycomb	Faces precured 2.5 hrs. @ 200°F sandwich 1 hr. @ 300°F, 1 hr. @ 330°F

TABLE 18 (CONT'D)
FABRICATION PROCESS FOR SANDWICH TEST SPECIMENS

Specimen Type	Specimen No.	Face Material Notes (2) (3) (4)	Core Material	Cure Cycle Note (1)
Compression Buckling Flat Panels (Cont'd)	241-6	Scotchply 1009	Aluminum Honeycomb	Faces precured 45 min. @ 330°F in press @ 250 p.s.i.; sandwich 2 hrs. @ 330°F
	241-7	181 Polyester Preimpregnated (5)	Glass Fabric Honeycomb	2 hrs @ 300°F
	241-8, 9, 10	181 Polyester Preimpregnated (5)	Polyurethane Foam	4 hrs. @ 235°F
	241-11	181 Polyester Preimpregnated (5)	Aluminum Multiwave	1 hr. @ 300°F, 1 hr @ 330°F
Compression Buckling Curved Panels	243-1, 2, 3	181 Polyester Preimpregnated	Aluminum Honeycomb	1 hr. @ 300°F, 1 hr. @ 330°F
	243-4, 5, 6	181 Epoxy Preimpregnated	Aluminum Honeycomb	2 hrs. @ 200°F, 1 hr. @ 250°F, 1 hr. @ 300°F
	243-8	181 Polyester Preimpregnated (5)	Aluminum Honeycomb	1 hr @ 300°F, 1 hr. @ 330°F
	243-9	181 Polyester Preimpregnated (5)	Aluminum Honeycomb	2 hrs.@ 200°F, 1 hr. @ 250°F, 1 hr. @ 300°F
	243-10, 11	181 Polyester Preimpregnated (5)	Aluminum Multiwave	1 hr. @ 300°F, 1 hr. @ 330°F
Flatwise Bending	245-1, 2	181 Polyester Preimpregnated (5)	Aluminum Honeycomb	1 hr. @ 300°F, 1 hr. @ 330°F

TABLE 18 (CONT'D)
FABRICATION PROCESS FOR SANDWITCH TEST SPECIMENS

Specimen Type	Specimen No.	Face Material Notes (2) (3) (4)	Core Material	Cure Cycle Note (1)
Flatwise Bending (Cont'd)	245-3	181 Epoxy Preimpregnated	Aluminum Honeycomb	2 hrs. @ 200°F, 1 hr. @ 250°F, 1 hr. @ 300°F
	245-4	181 Epoxy Preimpregnated (5)	Aluminum Honeycomb	Faces precured 2.5 hrs. @ 200°F sandwich 1 hr. @ 300°F, 1 hr. @ 330°F
	245-5, 6	181 Polyester Preimpregnated	Aluminum Honeycomb	1 hr. @ 300°F, 1 hr. @ 330°F
	245-7, 8	Scotchply 1009	Aluminum Honeycomb	Faces precured 45 min. @ 330°F sandwich 2 hrs. @ 330°F
	245-9, 10, 11	181 Polyester Preimpregnated (5)	Aluminum Multiwave	1 hr. @ 300°F, 1 hr. @ 330°F
	245-12, 13	181 Polyester Preimpregnated (5)	Glass Fabric Honeycomb	2 hrs. @ 300°F
	245-14, 15	181 Polyester Preimpregnated	Polyurethane Foam	4 hrs. @ 235°F

1. Cure pressure was by vacuum bag, 12-14 p.s.i., unless otherwise noted.
2. Type 181 glass cloth preimpregnated with polyester resin; Moboloy 81D, resin content 39%, Cordo Chemical Co.; or #PGLA, Resin content 43%, American Reinforced Plastics Co.
3. Type 181 glass cloth preimpregnated by Summit Industries with Shell Chemical Co. 828 epoxy resin and RP7A catalyst, resin content 38-42%.
4. Scotchply 1009 - Minnesota Mining and Manufacturing Co.
5. Adhesive used between faces and core, FM-97, Bloomingdale Rubber Co.

Compression Buckling

All aircraft structure is subject to reversible loadings and, therefore, must be designed for compression. The several designs based on studies for body structure, wings, empennages and control surfaces, used flat and curved sandwich structure, and curved laminated components as compression load carrying numbers. The analytical design of the structures utilized procedures outlined in Reference 38, MIL-HDBK-17, to determine allowable stresses. In order to substantiate the analysis, laboratory tests were made on flat sandwich panels, curved sandwich and curved laminates.

Flat Compression Panels

The compression buckling stress of flat sandwich panels of various types of constructions was determined. Variables investigated included face materials, core material, face thickness, and core thickness. The faces of all test specimens used 181 type cloth preimpregnated with polyester or epoxy resin. Core materials included aluminum honeycomb, fiberglass honeycomb, aluminum "Multiwave" and polyurethane foam.

All specimens were 22 inches square. The loaded edges were filled with Corefil 615, Bloomingdale Rubber Company, to prevent crushing. The tests were made with a Baldwin testing machine using a special fixture to distribute the load and to provide simple support for the edges of the specimen. The test setup is shown by Figure 43. The failing load was determined by applying load at a steady rate of 3000 to 4000 pounds per minute. Panel deflection was determined by measuring machine cross-head movement. Failing stresses were computed using a nominal face thickness of .010 per ply. A summary including descriptions of test specimens, failing load, mode of failure, failing stress and deflection is shown in Table 19. Most of the panels failed by local crippling rather than by panel buckling. Figure 44 shows a typical failure.

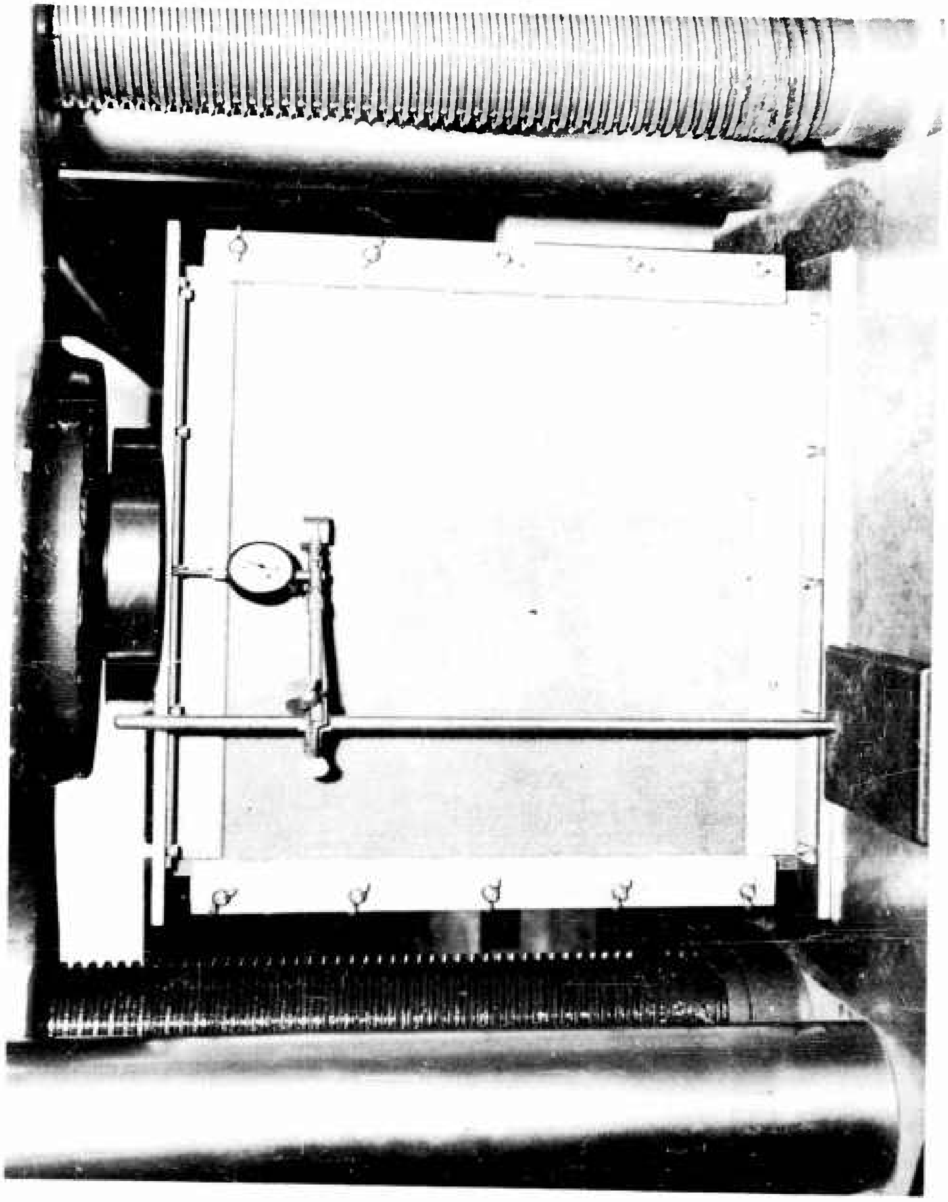


FIGURE 43. SETUP FOR COMPRESSION BUCKLING
TESTS OF FLAT SANDWICH PANELS

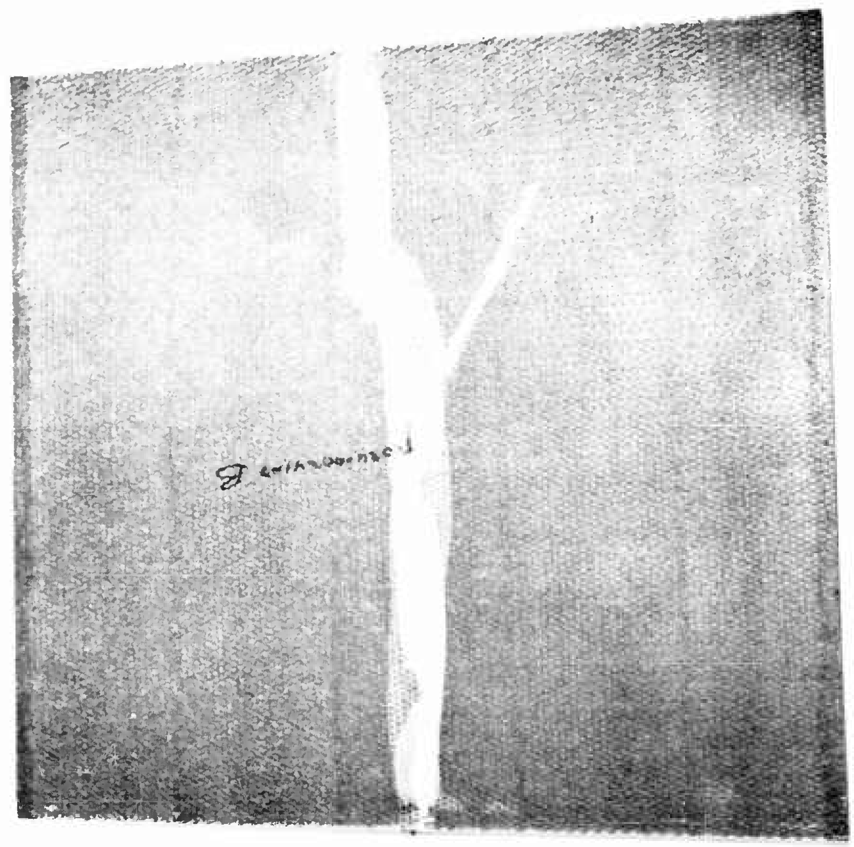


FIGURE 44. TYPICAL COMPRESSION BUCKLING FAILURE OF FLAT SANDWICH PANEL

TABLE 19

COMPRESSIVE BUCKLING TESTS OF FLAT SANDWICH PANELS

Specimen Number	Face Material	Core Material	No. Plies	t_c (in.)	Max. Load (lb.)	Average Face Stress At Failure (p.s.i.)	Type Failure (3)
241-1A	181 Cloth Polyester Resin (1)	AL-1/8-3003-.0007P	2	.75	22450	25520	A
-1B	"	"	2	.75	20700	23550	A
-1C	"	"	2	.75	23700	26920	A
241-2A	"	"	2	.5	13450	15280	A
-2B	"	"	2	.5	11850	13470	A
-2C	"	"	2	.5	11400	12960	A
-2D	" (2)	"	2	.5	23500	26700	A
-2E	"	"	2	.5	27500	31200	A
-2F	"	"	2	.5	23500	26700	A
241-3A	" (1)	"	3	.5	18600	14090	B
-3B	"	"	3	-	-	-	-
-3C	"	"	3	.5	16900	12800	B
241-4A	"	"	5	.5	26050	11840	B
-4B	"	"	5	.5	24100	10950	A
-4C	"	"	5	.5	23900	10860	A
-4D	" (2)	"	5	.5	38650	17500	A
-4E	"	"	5	.5	51300	23350	A
-4F	"	"	5	.5	40650	18500	A
241-5A	181 Cloth Epoxy Resin	"	2	.5	25000	28410	A
-5B	"	"	2	.5	25350	11520	A
-5C	"	"	2	.5	25150	11430	A
241-7A	181 Cloth Polyester Resin	-	3	.5	27800	21060	A
-7B	"	"	3	.5	27700	20980	A

TABLE 19 (CONT'D)
COMPRESSIVE BUCKLING TESTS OF FLAT SANDWITCH PANELS

Specimen Number	Face Material	Core Material	No. Plies	t_c (in.)	Max. Load (lb.)	Average Face Stress At Failure (p.s.i.)	Type Failure (3)
241-7C	181 Cloth Polyester Resin (1)	AL-1/8-3003-.0007P	3	.5	21900	16590	A
241-8A	"	Polyurethane Foam 3 lb./ft.^{3}	3	.5	3750	2840	C
-8B	"	"	3	.5	3550	2690	C
-8C	"	"	3	.5	5380	4080	C
241-9A	"	"	3	1.0	5420	4110	B
-9B	"	"	3	1.0	5120	3880	C
-9C	"	"	3	1.0	6875	5210	C
241-10A	"	Polyurethane Foam 4.5 lb./ft.^{3}	3	.5	10865	8230	C
-10B	"	"	3	.5	10800	8180	C
-10C	"	"	3	.5	9650	7310	C
241-11A	"	Al. Multiwave 4.5 lb/ft.^{3}	3	.5	17600	13330	A
-11B	"	"	3	.5	17350	13140	A
-11C	"	"	3	.5	18300	13860	A

(1) Cordo Chemical Company 81D
(2) American Reinforced Plastics #PGLA
(3) Letter code for Type Failure Column:
A. Compressive failure of face
B. Failure of bond between core and faces
C. Buckling of entire panel

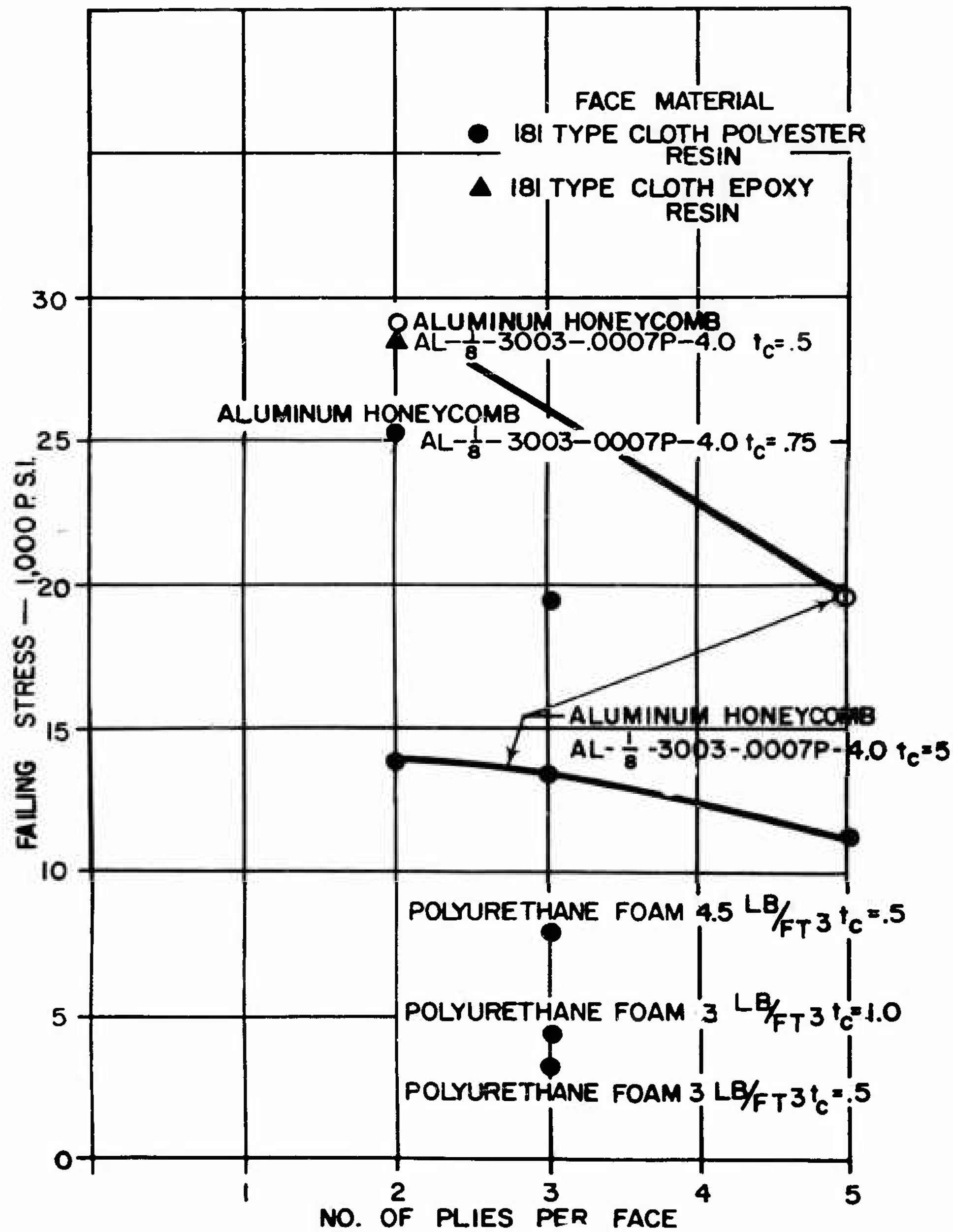


FIGURE 45. FAILING STRESS FOR FLAT 22 X 22 INCH SANDWICH PANELS OF VARIOUS FACE AND CORE MATERIALS

Curved Compression Panels

Two types of curved panels were investigated to determine the compression buckling strength. Curved laminates were tested to substantiate the analytical studies for the H-23 type tail boom structure. Tests of curved sandwich panels were made to substantiate the studies for the HU-1 tail boom. All laminates and sandwich faces were made with type 181 glass cloth preimpregnated with polyester or epoxy resin. Thickness, resin, and radius were the variables for the laminated specimens. In the sandwich panels, the effects of core materials and thickness were investigated in addition to the same variables for laminates. All specimens are 90 degree segments of a cylinder 24 inches long. The loaded edges of the sandwich panels were filled with Corfil 615, Bloomingdale Rubber Company, to prevent crushing.

Tests were made in a Baldwin testing machine using a special fixture to distribute the load and to provide simple edge support to the unloaded edges. The test setup is shown in Figure 46. The failing load was determined by applying load at a steady rate of 2000 to 4000 pounds per minute. Panel deflection was determined by measuring cross-head movement. Failing stresses were computed using a nominal thickness of .010 inch per ply of glass cloth. A summary including descriptions of this test specimen's failing load, mode of failure, failing stress and deflection is shown in Table 20. Typical failures are shown in Figures 47 and 48. The strain at failure was determined by measuring cross-head movement and is not considerable reliable.

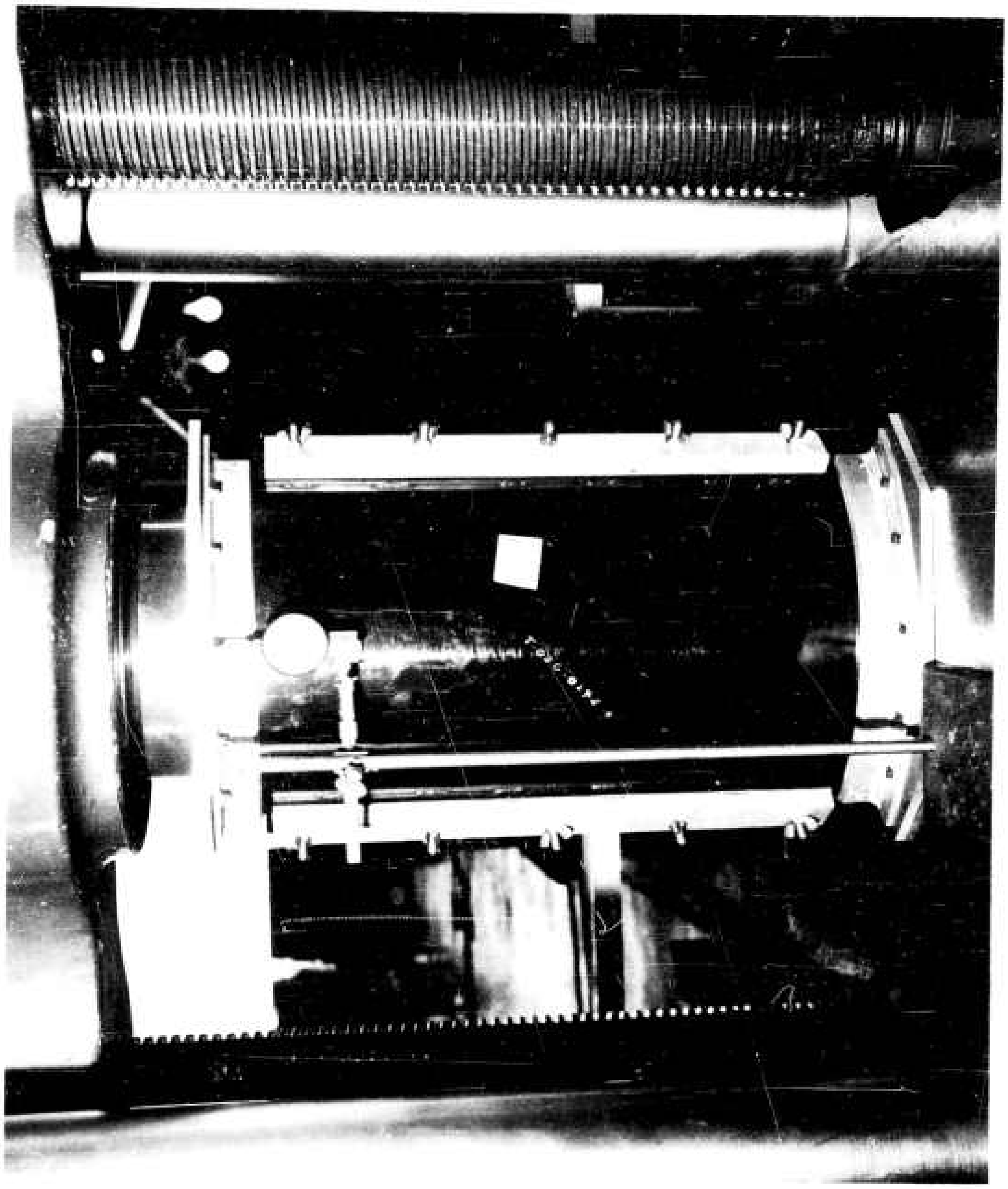


FIGURE 46. SETUP FOR COMPRESSION BUCKLING TESTS OF CURVED PANELS

FIGURE 47. TYPICAL COMPRESSION FAILURE OF CURVED LAMINATE PANEL

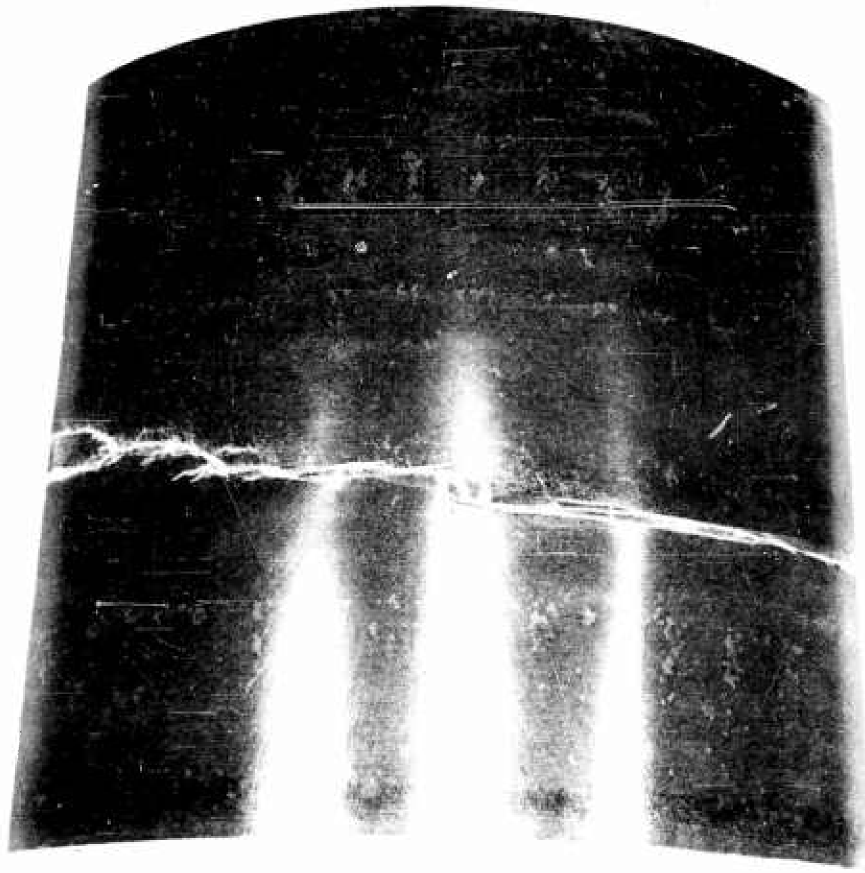


FIGURE 48. TYPICAL COMPRESSION FAILURE OF CURVED SANDWICH PANEL

TABLE 20
COMPRESSIVE BUCKLING OF CURVED LAMINATED PANELS

Specimen Number	Face Material	No. Plies	Inside Radius (in.)	Max. Load (lb.)	Max. Stress (p.s.i.)	Type* Failure	Strain At Failure (in./in. x 10^{-3})
242-1A	181 Cloth Polyester Resin	5	6	3440	7300	A	2.83
-1B	"	5	6	2800	5950	A	3.12
-1C	"	5	6	3280	6960	A	2.62
242-2A	"	9	6	13000	15330	B	5.030
-2B	"	9	6	13650	16100	B	5.190
-2C	"	9	6	12000	14150	B	5.890
242-3A	"	9	9	12600	9910	A	3.780
-3B	"	9	9	12500	9830	A	3.285
-3C	"	9	9	11050	8690	A	3.740
242-4A	"	12	9	21100	12440	A	4.870
-4B	"	12	9	25400	14980	B	5.900
-4C	"	12	9	22700	13380	B	5.630
242-5A	181 Cloth Epoxy Resin	5	6	3200	6790	A	3.20
-5B	"	5	6	2390	5070	A	2.66
-5C	"	5	6	2910	6180	A	2.90
242-6A	"	9	6	8550	10080	A	4.49
-6B	"	9	6	11480	13540	B	5.11
-6C	"	9	6	8000	9430	A	3.66
242-7A	"	9	9	12100	9510	B	3.19
-7B	"	9	9	11700	9200	B	3.74
-7C	"	9	9	10800	8490	A	4.49
242-8A	"	12	9	25600	15090	A	5.49
-8B	"	12	9	23250	13710	A	4.49
-8C	"	12	9	23100	13620	A	4.16

TABLE 20 (CONT'D)
COMPRESSIVE BUCKLING OF CURVED LAMINATED PANELS

Specimen Number	Face Material	No. Plies	Inside Radius (in.)	Max. Load (lb.)	Max. Stress (p.s.i.)	Type* Failure	Strain At Failure (in./in. x 10^{-3})
242-10A	20 End Roving Epoxy	12	9	13700	8040	A	3.16
-10B	"	12	9	11550	6770	A	2.70
-10C	"	12	9	8400	4930	A	3.24
242-11A	"	15	9	18700	8750	A	2.75
-11B	"	15	9	18100	8470	A	3.66
-11C	"	15	9	11570	5420	A	2.08

*Letter code for Type Failure Column: A. Local buckling; B. Compressive failure or cracking of laminate.

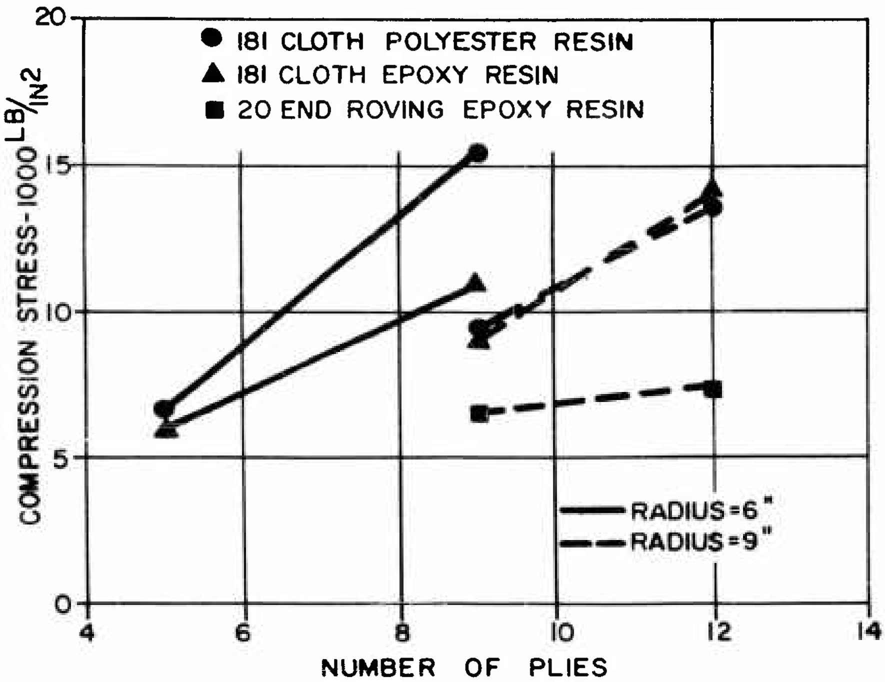


FIGURE 49. COMPRESSION BUCKLING STRESS OF CURVED LAMINATED PANELS

TABLE 21
COMPRESSIVE BUCKLING TEST OF CURVED SANDWICH PANELS

Specimen Number	Face Material	Core Material	No. Of Plies	t_c (in.)	Inside Radius (in.)	Max. Load (lb.)	Max. Stress (p.s.i.)	Type Failure (3)	Strain At Failure in./in.
243-1A	181 Cloth Polyester Resin (2)	AL-1/8-3003-.0007P	2	.375	9	15900	27510	A	.0072
-1B	"	"	2	.375	9	14800	25560	A	.0066
-1C	"	"	2	.375	9	14800	25610	A	.0063
243-2A	"	"	2	-	16	-	-	-	-
-2B	"	"	2	.375	16	22600	22200	A	.0057
-2C	"	"	2	.375	16	24000	23580	A	.0064
243-3A	" (1)	"	3	.375	16	27150	17790	A	.0055
-3B	"	"	3	.375	16	23600	15450	A	.0044
-3C	"	"	3	.375	16	24350	15940	A	.0058
243-4A	181 Cloth Epoxy Resin	"	2	.375	9	10150	17560	A	.0073
-4B	"	"	2	.375	9	21300	36850	A	.0108
-4C	"	"	2	.375	9	19350	33480	A	.0098
243-5A	"	"	2	.375	16	40350	39640	A	.0097
-5B	"	"	2	.375	16	36350	36030	A	.0101
-5C	"	"	2	.375	16	40150	39440	A	.0101
243-6A	"	"	4	.375	16	45930	22540	B	.0059
-6B	"	"	4	.375	16	42500	20850	B	.0062
-6C	"	"	4	.375	16	40000	19630	A	.0051
243-8A	181 Cloth Polyester Resin (1)	"	2	.188	9	10500(4)	18350	B	.0064
-8B	"	"	2	.188	9	14900	26050	A	.0072
-8C	"	"	2	.188	9	14450	25260	A	.0079
243-9A	181 Cloth Epoxy Resin	"	2	.188	9	21800	38060	A	.0119

TABLE 21 (CONT'D)
COMPRESSIVE BUCKLING TEST OF CURVED SANDWICH PANELS

Specimen Number	Face Material	Core Material	No. Of Plies	t_c (in.)	Inside Radius (in.)	Max. Load (lb.)	Max. Stress (p.s.i.)	Type Failure (3)	Strain At Failure in./in.
243-9B	181 Cloth Epoxy Resin	AL-1/8-3003-.0007P	2	.188	9	23500	41010	A	.0120
-9C	"	"	2	.188	9	22550	39420	A	.0121
243-10A	"	Aluminum Multiwave 4.5 lb./ft.^{3}	2	.375	9	8250	14270	A	.0069
-10B	"	"	2	.375	9	9000	15570	A	.0052
-10C	"	"	2	.375	9	9500	16440	A	.0053
-10D	"(2)	"	2	.375	9	16300	28250	B	.0071
-10E	"(2)	"	2	.375	9	13700	23710	A	.0070
-10F	"(2)	"	2	.375	9	13150	22800	A	.0008
243-11A	"	"	2	.188	6	6520	16850	-	.0053
-11B	"	"	2	.188	6	6090	15700	-	.0066
-11C	"	"	2	.188	6	4565	11800	-	.0058

(1) Cardo Chemical Company Moboloy 81D
(2) American Reinforced Plastics Company No. PGLA
(3) Letter code for Type of Failure Column:
A. Compressive face failure.
B. Crushing at end.
(4) This panel was out of square and considered defective.

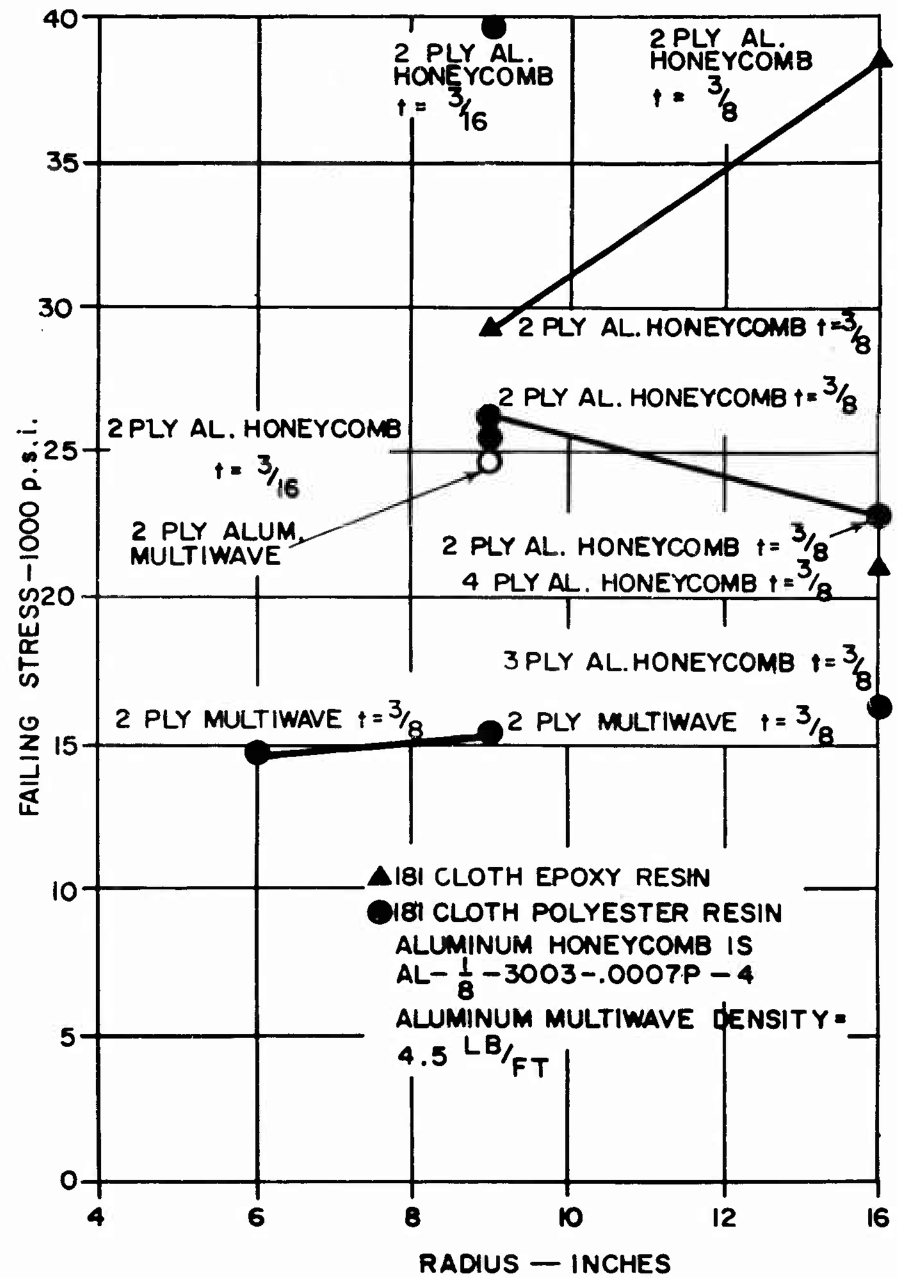


FIGURE 50. COMPRESSION BUCKLING STRESS OF CURVED SANDWICH PANELS

PANEL SHEAR

All aircraft structural components are in some manner loaded in shear. Very little information on panel shear strength of reinforced plastics is available for use in design analysis. A comprehensive test program to evaluate all variables would require considerable effort and was beyond the scope of this contract. However, some basic information was needed and the tests in this program were confined to an investigation of the comparative effects of various materials and thicknesses for one size panel.

The laminates and the faces of the sandwich panels were made with type 181 glass cloth preimpregnated with polyester or epoxy resins. Core materials for sandwich panels included aluminum honeycomb, aluminum Multiwave, and polyurethane foam. All panels were 16 inches square. The edges of the sandwich panels were filled with Corfil 615, Bloomingdale Rubber Company, to facilitate attachment of the panel to the test fixture, and aluminum alloy strips were bonded to all specimens to increase the bearing strength for the edge attachment. The test setup using the special test fixture is shown in Figure 51. The pivot points are 15 inches apart, and the diagonal distance between the upper and lower loading points is 21.21 inches. The panels were loaded to failure by applying tension load to the fixture, resulting in pure shear stresses in the panel. Some of the panels failed in skin shear, some by shear failure between the attachments, and others in the bond between the face and core. Usually these failures were combined and practically instantaneous, making it impossible to determine which failure occurred first.

Table 22 is a summary of test results showing specimen description, failing load, failing stress, deflection and computed modulus of rigidity. The failing stresses were determined by using a laminate and face thickness of .010 per ply of 181 cloth. Deflection was determined by measuring head movement. Typical failures of the panels are shown in Figures 52 and 53.

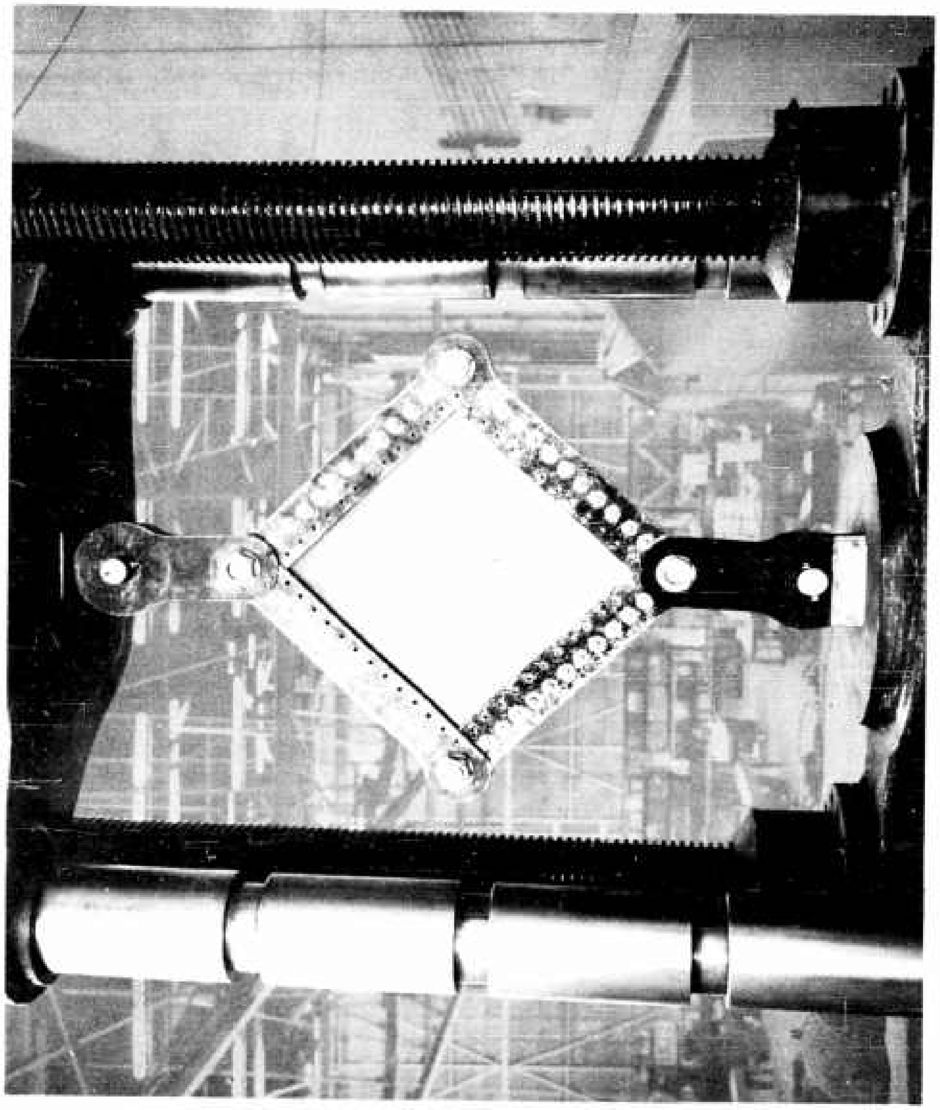


FIGURE 51. TEST SETUP FOR PANEL SHEAR

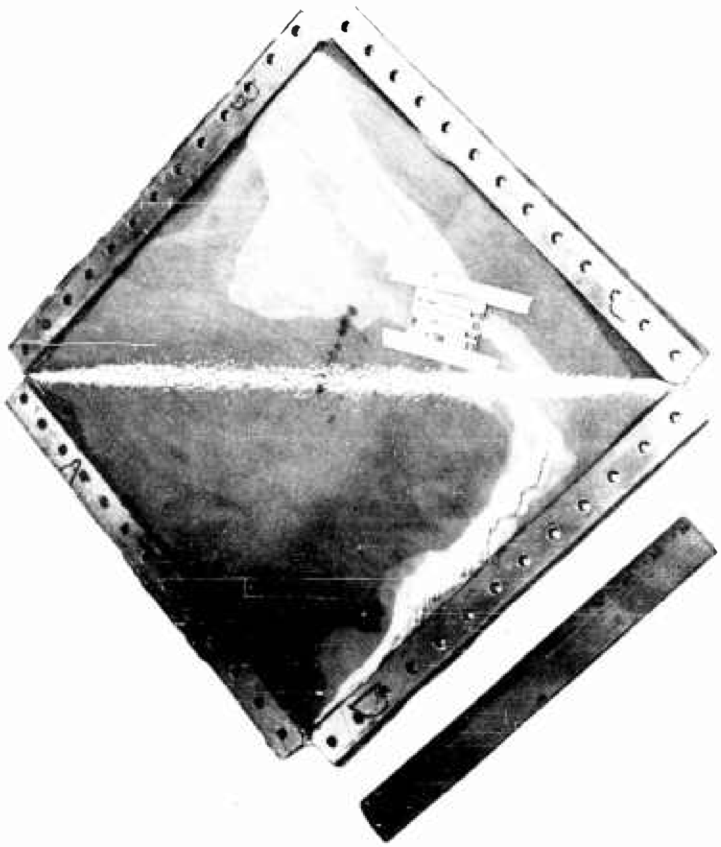


FIGURE 52. TYPICAL FAILURE OF FLAT LAMINATED SHEAR PANEL

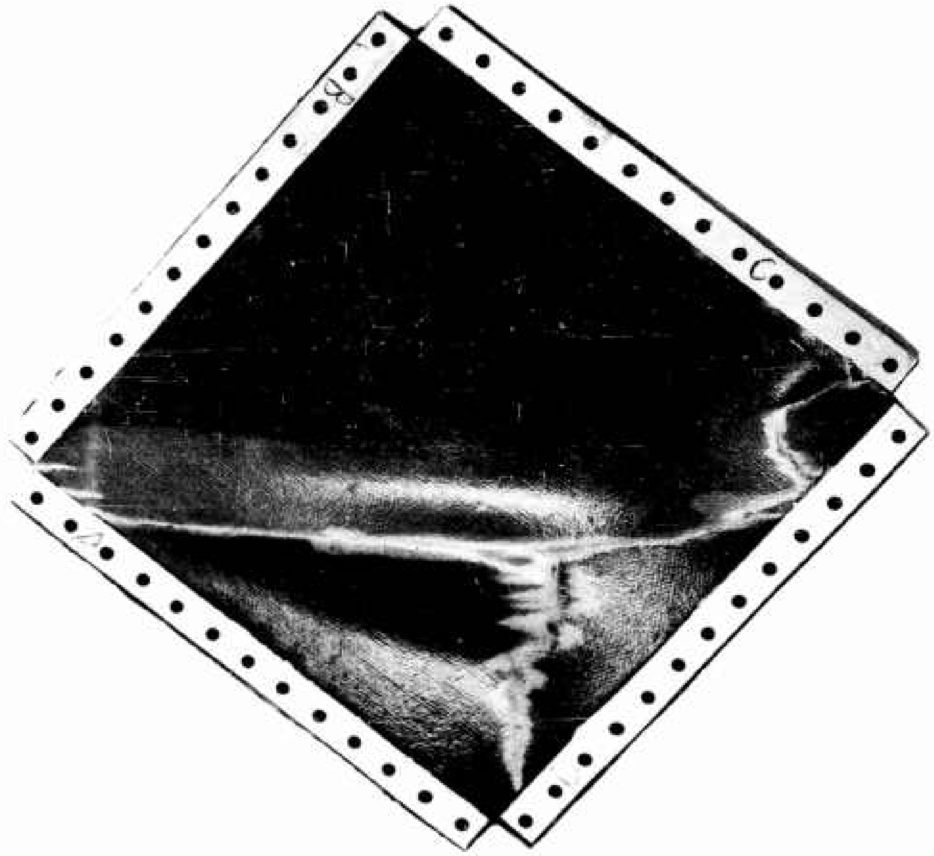


FIGURE 53. TYPICAL FAILURE OF FLAT SANDWICH SHEAR PANEL

TABLE 22
SHEAR TESTS OF FLAT LAMINATED PANELS

Specimen Number	Face Material	No. Of Plies	Ultimate Load (lb.)	Maximum Stress (p.s.i.)	Buck Visible (p.s.i.)	Total* Strain (in.)
239-1A	181 Cloth Polyester Resin	4	13800	16260	1130	.585
-1B	"	4	11300	13320	2560	.401
-1C	"	4	13200	15560	-	.531
239-2A	"	8	26100	15380	3900	.562
-2B	"	8	24350	14350	2315	.579
-2C	"	8	26200	15440	2960	.527
239-3A	"	12	24150	9490	1613	.430
-3B	"	12	35500	13950	3340	.530
-3C	"	12	24100	9470	3250	.362
239-4A	181 Cloth Epoxy Resin	4	10180	12000	-	.300
-4B	"	4	8920	10520	821	.272
-4C	"	4	8940	10540	1609	.287
239-5A	"	8	18200	10720	2270	.291
-5B	"	8	19150	11280	2785	.322
-5C	"	8	17200	10140	2210	.330
239-6A	"	12	51000	20040	5990	-
-6B	"	12	49950	19630	4040	.596
-6C	"	12	47900	18820	4110	.517
239-7A	1009 Scotchply Crossplied	8	20500	12080	3590	1.836
-7B	"	8	19600	11550	3240	1.470
-7C	"	8	21450	12640	4280	1.650
239-8A	1009 Scotchply Isotropic	8 or 9	30000	15720	4540	.485
-8B	"	8 or 9	32650	17100	5070	.592
-8C	"	8 or 9	28500	14930	5330	.503

*Total strain measured over diagonal distance of 21.21 inches using cross-head movement.

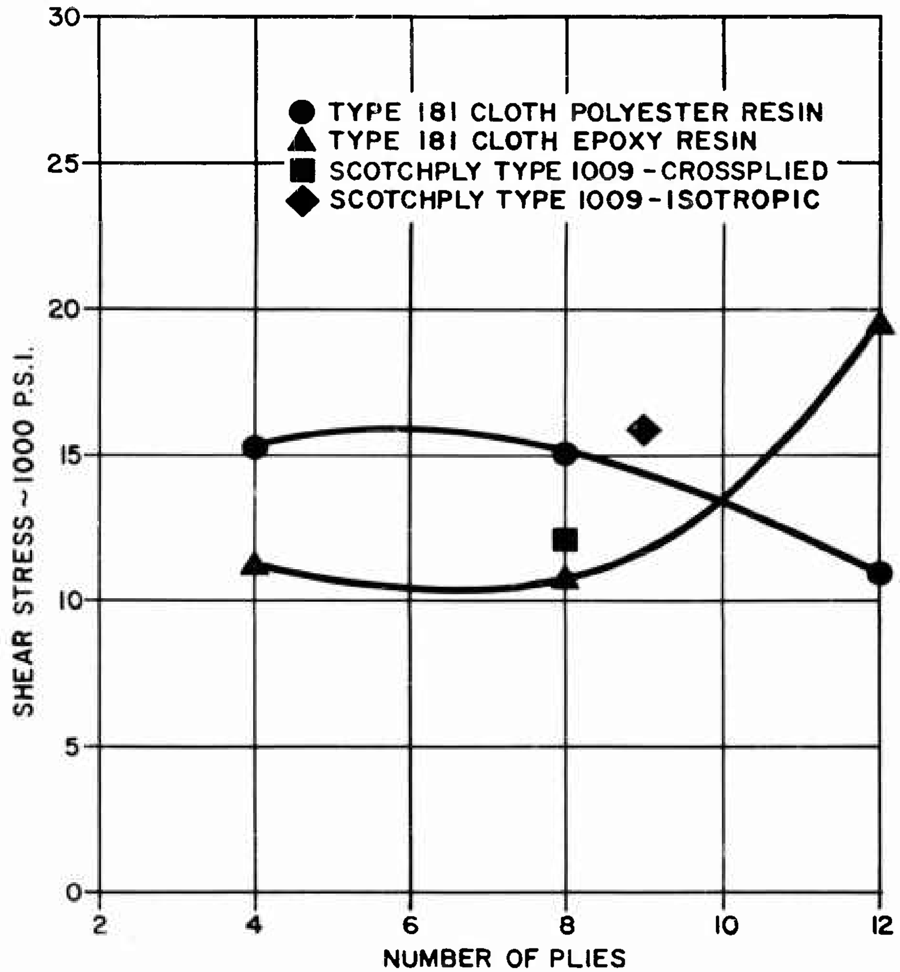


FIGURE 54. SHEAR FAILURE STRESS OF 16 X 16 INCH LAMINATED PANELS

TABLE 23
SHEAR TESTS OF FLAT SANDWICH PANELS

Specimen No.	Face Material	Core Material	No. Of Plies	Core Thickness (in.)	Ultimate Load (p.s.i.)	Maximum Stress (p.s.i.)	Total * Strain (in.)
240-1A	181 Cloth Polyester Resin (1)	AL-1/8-3003-.0007P	2	.375	18500	21800	.404
-1B	"	"	2	.375	18150	21390	.630
-1C	"	"	2	.375	19400	22860	.413
-1D	" (2)	"	2	.375	16700	19650	.389
-1E	"	"	2	.375	16050	18900	.330
-1F	"	"	2	.375	17200	20250	.403
240-2A	" (1)	"	3	.375	31500	24750	.510
-2B	"	"	3	.375	34700	27260	.604
-2C	"	"	3	.375	31400	24670	.549
240-3A	"	"	5	.375	34000	16030	.436
-3B	"	"	5	.375	34600	16310	.554
-3C	"	"	5	.375	33500	15790	.498
240-4A	181 Cloth Epoxy Resin	"	2	.375	28300	33350	.528
-4B	"	"	2	.359	24900	29340	.413
-4C	"	"	2	.375	27000	31820	.540
240-5A	"	"	3	.375	38000	29850	.442
-5B	"	"	3	.375	31000	24350	.648
-5C	"	"	3	.375	36200	28440	.604
240-6A	"	"	5	.375	36750	17320	.414
-6B	"	"	5	.375	32950	15530	.460
-6C	"	"	5	.375	46200	21780	.475
240-7A	1009 Scotch-ply	"	3	.375	64250	50470	.642
-7B	"	"	3	.375	15000	11780	.308
-7C	"	"	3	.375	59000	46348	.750

TABLE 23 (CONT'D)
SHEAR TESTS OF FLAT SANDWICH PANELS

Specimen No.	Face Material	Core Material	No. Of Plies	Core Thickness (in.)	Ultimate Load (p.s.i.)	Maximum Stress (p.s.i.)	Total * Strain (in.)
240-8A	181 Cloth Polyester Resin (2)	AL-1/8-3003-.0007P	2	.375	13900	16400	.288
-8B	"	"	2	.375	13200	15600	.233
-8C	"	"	2	.375	13650	16120	.450
240-9A	"	"	3	.375	21250	16700	.483
-9B	"	"	3	.375	20800	16350	.476
-9C	"	"	3	.375	22350	17550	.270
-9D	" (2)	"	3	.375	17500	13750	.726
-9E	"	"	3	.375	18150	14250	.850
-9F	"	"	3	.375	17900	14050	.870
240-10A	"	"	5	.375	41700	19660	.660
-10B	"	"	5	.375	41150	19400	.598
-10C	"	"	5	.375	43550	20430	.655
240-11A	"	"	2	.188	18200	21450	.510
-11B	"	"	2	.188	15800	18620	.434
-11C	"	"	2	.188	-	-	-
240-12A	"	"	3	.188	25500	20030	.462
-12B	"	"	3	.188	9000	-	-
-12C	"	"	3	.188	-	-	-
240-13A	"	Polyurethane Foam 3 lb./ft.^{3}	2	.75	18500	21800	.670
-13B	"	"	2	.75	14500	17090	.530
-13C	"	"	2	.75	18500	21800	.728
240-14A	"	Polyurethane Foam 4.5 lb./ft.^{3}	2	.750	19800	23330	.474
-14B	"	"	2	.750	19250	22680	.459
-14C	"	"	2	.750	19250	22680	.535

TABLE 23 (CONT'D)
SHEAR TESTS OF FLAT SANDWICH PANELS

Specimen No.	Face Material	Core Material	No. Of Plies	Core Thickness (in.)	Ultimate Load (p.s.i.)	Maximum Stress (p.s.i.)	Total * Strain (in.)
240-15A	181 Cloth Polyester Resin (1)	Polyurethane Foam 4.5 lb./ft.^{3}	2	.375	11000	12960	.333
-15B	"	"	2	.375	11000	12960	.370
-15C	"	"	2	.375	8700	10250	.325

* Total strain measured over diagonal distance of 21.21 inches using cross-head movement.

(1) Cordo Chemical Company, (Moboloy 81D)
(2) American Reinforced Plastics Co., No. PGLA

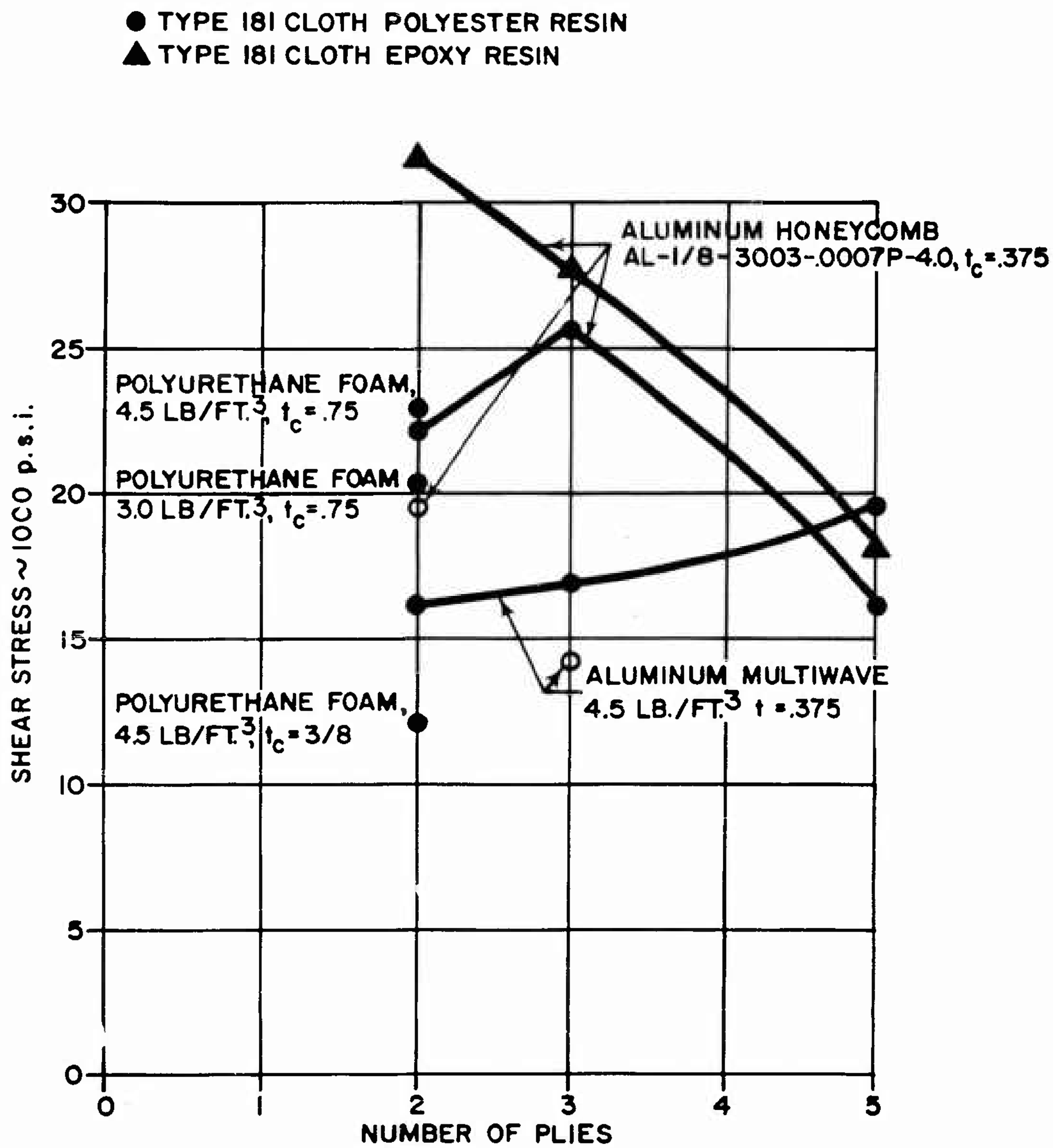


FIGURE 55. SHEAR FAILURE STRESS OF 16 X 16 INCH SANDWICH PANELS

BENDING OF FLAT SANDWICH PANELS

Body, wing, and other similar structure of sandwich construction will be subjected to panel bending loads from airload and miscellaneous local loads imposed on the structure. Tests were made of flat panels loaded as simple beams to determine the effects of face thickness and core material.

The faces of all panels were made with type 181 glass cloth preimpregnated with polyester or epoxy resin. Core materials investigated included aluminum honeycomb, aluminum "Multiwave", fiberglass honeycomb and polyurethane foam. All panels were 36 inches long, 4 inches wide and with a core 1.0 inch thick. The beams were loaded at two points 17 inches apart and equidistant from the support points, which were 34 inches apart. The load application and support points had a radius of 3/4 inch. A photograph of the test setup is shown in Figure 56.

Load was applied at a steady rate of 120 to 200 pounds per minute until the panel failed. Table 24 is a summary of panel description, failing load, failing stress, mode of failure, the deflection. The failing stresses were computed using a nominal thickness of .010 inch per ply.

The test results are rather erratic and are not conclusive. This is believed to be due primarily to the poor bonds between the faces and cores, which were not discovered until these and other tests were run. Many of the specimens failed in shear in this bond. Figure 53 shows typical failures of specimens that failed in face compression.

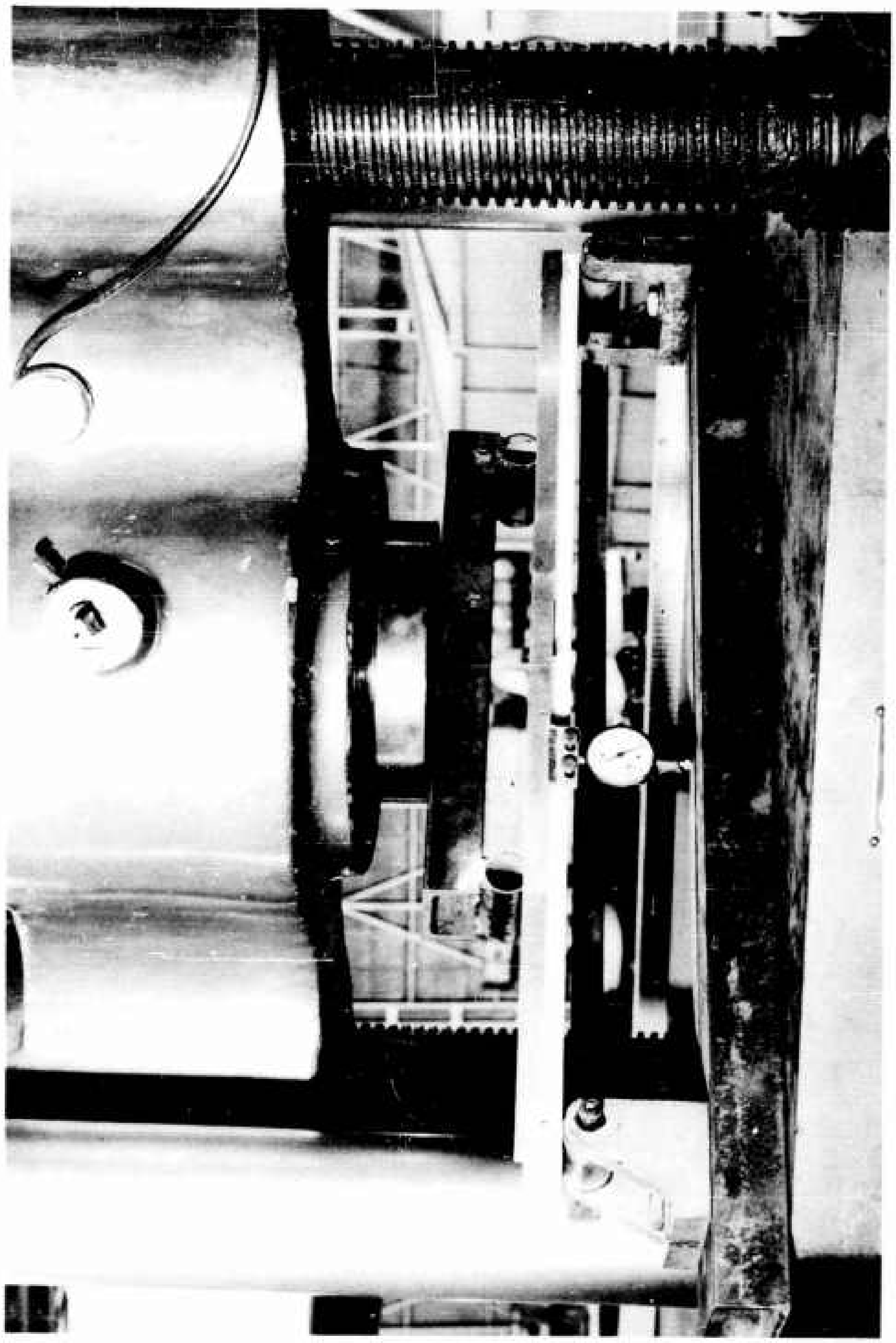


FIGURE 56. SETUP FOR BENDING TESTS OF FLAT SANDWICH PANELS

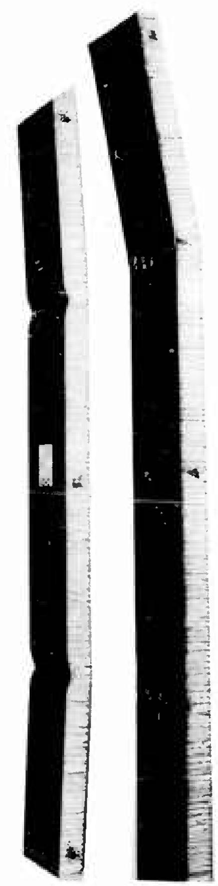


FIGURE 57. TYPICAL BENDING FAILURES OF FLAT SANDWICH PANELS

TABLE 24

BENDING TEST FOR FLAT SANDWICH PANELS

Specimen Number	Face Material	Core Material (4)	No. Of Plies	Ribbon Direction	Max. Load (lb.)	Max. Deflection (in.)	Average Failing Stress (p.s.i.)	Type Failure (3)
245-1A	181 Cloth Polyester Resin (1)	AL-1/8-3003-.0007P	2	Long.	430	1.505	22400	A
-1B	"	"	2	Long.	450	1.590	23430	A
-1C	"	"	2	Long.	382	1.375	19900	A
-1D	"(2)	"	2	Long.	590	2.057	30750	A
-1E	"	"	2	Long.	550	2.100	28700	A
-1F	"	"	2	Long.	594	2.167	30950	A
245-2A	"(2)	"	3	Long.	850	1.930	29300	A
-2B	"(2)	"	3	Long.	860	1.925	29600	A
-2C	"(2)	"	3	Long.	660	1.533	22700	A
245-3A	181 Cloth Epoxy Resin	"	2	Long.	945	3.060	49220	B
-3B	"	"	2	Long.	780	2.045	40630	B
-3C	"	"	2	Long.	972	2.750	50630	B
245-4A	"	"	3	Long.	754	1.615	25920	B
-4B	"	"	3	Long.	820	1.772	28200	B
-4C	"	"	3	Long.	696	1.492	23930	C
245-5A	181 Cloth Polyester Resin (1)	"	2	Lat.	320	1.090	16670	A
-5B	"	"	2	Lat.	496	1.770	25830	A
-5C	"	"	2	Lat.	484	1.740	25210	A
245-6A	"	"	3	Lat.	662	1.675	22770	A
-6B	"	"	3	Lat.	804	1.767	27650	A
-6C	"	"	3	Lat.	666	1.677	22900	A
245-7A	Scotchply 1009 Crossply	"	2	Long.	402	1.245	20930	E

TABLE 24 (CONT'D)
BENDING TEST FOR FLAT SANDWICH PANELS

Specimen Number	Face Material	Core Material (4)	No. Of Plies	Ribbon Direction	Max. Load (lb.)	Max. Deflection (in.)	Average Failing Stress (p.s.i.)	Type Failure (3)
245-7B	Scotchply 1009 Crossply	AL-1/8-3003-.0007P	2	Long.	820	1.770	42710	B
-7C	"	"	2	Long.	604	1.275	31460	B
245-8A	Scotchply 1009 Isotropic	"	3	Long.	1492	2.130	51300	B
-8B	"	"	3	Long.	1560	2.055	51210	B
-8C	"	"	3	Long.	1500	2.587	51580	B
245-9A	181 Cloth Polyester Resin (1)	Aluminum Multiwave 4.5 lb./ft.^{3}	2	Long.	366	1.844	19070	A
-9B	"	"	2	Long.	376	1.424	19580	A
-9C	"	"	2	Long.	426	1.760	22180	A
245-10A	"	"	3	Long.	712	1.870	24480	B
-10B	"	"	3	Long.	804	2.092	27650	A
-10C	"	"	3	Long.	685	1.830	23550	A
-10D	" (2)	"	3	Long.	696	2.158	24000	B
-10E	"	"	3	Long.	804	2.407	27650	B
-10F	" (1)	"	3	Long.	670	1.950	23050	B
245-11A	"	"	5	Long.	1350	2.187	27320	A
-11B	"	"	5	Long.	1320	2.180	26710	A
-11C	"	"	5	Long.	1260	2.043	25500	A
245-12A	"	NP-3/16-112-4.5	2	Long.	494	2.097	25740	B
-12B	"	"	2	Long.	470	1.967	24490	B
-12C	"	"	2	Long.	336	1.387	17500	A
245-13A	"	"	3	Long.	329	.905	11310	A
-13B	"	"	3	Long.	492	1.455	16920	A
-13C	"	"	3	Long.	680	1.830	23380	A

TABLE 24 (CONT'D)
BENDING TEST FOR FLAT SANDWICH PANELS

Specimen Number	Face Material	Core Material (4)	No. Of Plies	Ribbon Direction	Max. Load (lb.)	Max. Deflection (in.)	Average Failing Stress (p.s.i.)	Type Failure (3)
245-14A	181 Cloth Polyester Resin (1)	Polyurethane Foam 3.0 lb./ft.^{3}	2	-	104	.625	5420	D
-14B	"	"	2	-	100	.595	5210	D
-14C	"	"	2	-	85	.650	4420	D
245-15A	"	"	3	-	124	.565	4260	D
-15B	"	"	3	-	126	.712	4340	D
-15C	"	"	3	-	95	.812	3270	D

(1) Cordo Chemical Company Moboloy 81D
(2) American Reinforced Plastics Company No. PGLA
(3) Letter indicates type of failure (See Below):
A. Failure of bond between face and core.
B. Compressive failure of upper face.
C. Core shear.
D. Depression of core at load points.
E. Tensile failure of lower face.
(4) t_c for all specimens = 1.0 inch.

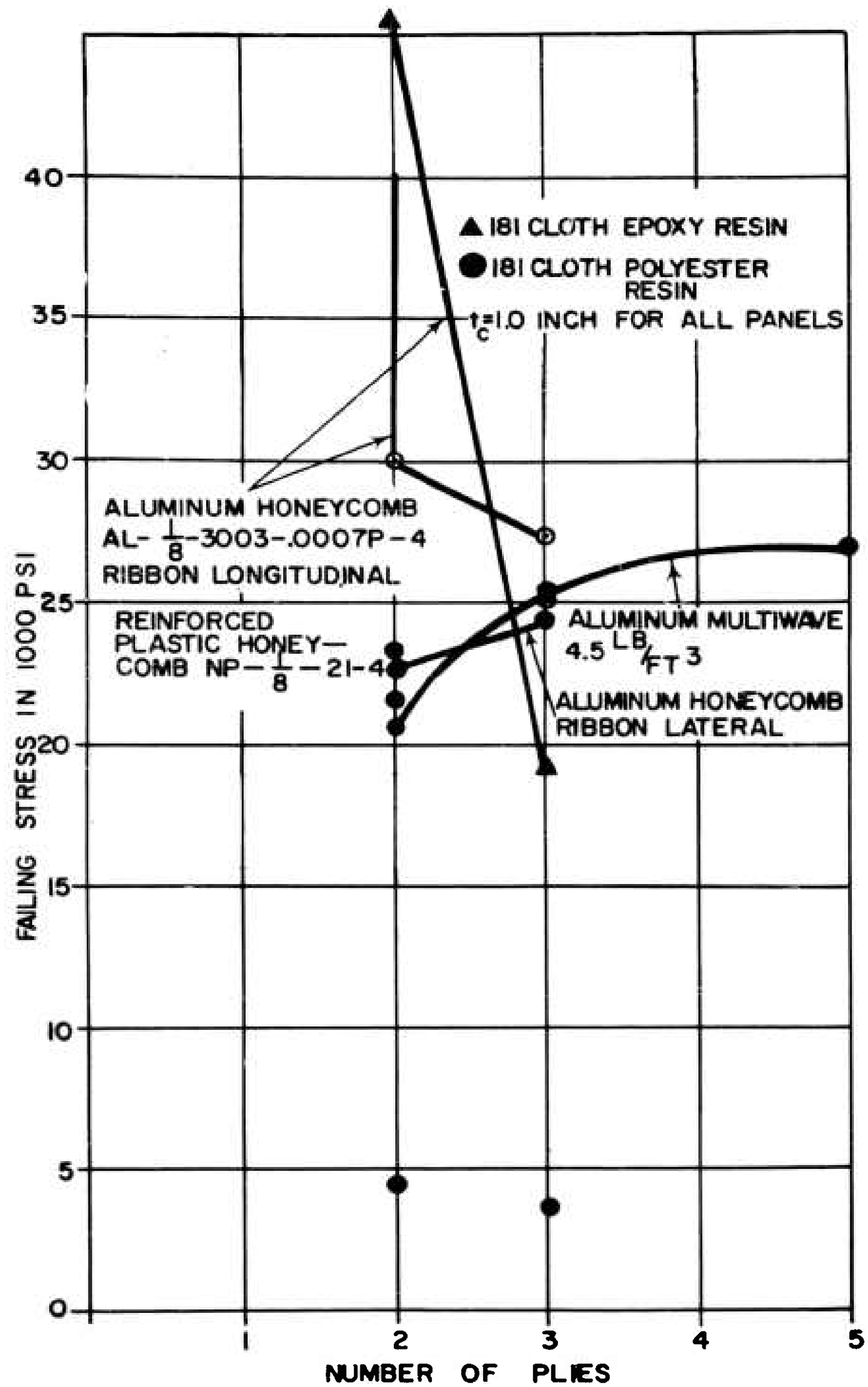


FIGURE 58. BENDING FAILURE STRESS OF SANDWICH PANELS

FASTENERS IN SOLID LAMINATES

Very little test data are readily available on the strength of mechanical fasteners in reinforced plastics and of adhesive bonded joints. Reference 38 , MIL-HDBK-17, contains some information on strength of fasteners and is considered to be a good guide. A complete, comprehensive test program, which is highly desirable, was beyond the scope of work of this contract; however, some testing was considered to be essential.

Tests were accomplished in the Hayes laboratory to determine failing strength of bolted and riveted attachments in type 181 fiberglass cloth preimpregnated with polyester resin and epoxy resin, and with a nonwoven epoxy resin impregnated fabric, "Scotchply". Tests of adhesive bonded joints using reinforced plastic test specimens supplied by Hayes were bonded and tested by Bloomingdale Rubber Company and by Minnesota Mining and Manufacturing Company.

BOLTED AND RIVETED JOINTS

Tests were made to determine the failing strength of single attachment riveted and bolted joints in various solid laminates. The fasteners and materials that were investigated are listed below. Some of the bolted joints included laminates with molded-in aluminum strips to increase the bearing strength.

Fasteners

1. Flush-head rivets, type MS 20426AD, with diameters of 1/8, 5/32, and 3/16 inch.

2. Protruding-head rivets, type MS 20470AD, with diameters of 1/8, 5/32, and 3/16 inch.

3. Flush-head screws, type AN 509, with diameters of 3/16 and 1/4 inch.

4. Protruding-head bolts, type AN-3 and AN-4.

Materials

1. Type 181 glass cloth impregnated with epoxy resin.

2. Type 181 glass cloth impregnated with polyester resin.

3. Type 1002 Scothply with cross-plied fiber orientation (alternating plies having fibers at 90°).

4. Type 1002 Scothcply with isotropic fiber orientation (fibers of adjacent plies oriented 60° apart; used in multiples of 3).

The laminated test specimens were 2 inches wide and 6 inches long. The fastener was located on the centerline at an edge distance from the end equal to twice the diameter of the fastener. In specimens using metal inserts, two-thirds of the numbers of plies were on the side of the insert toward the head of the fastener and one-third of the opposite side. Fabric warp orientation was parallel to the longitudinal axis for type 181 cloth, parallel and perpendicular for cross-ply Scothply, and 0^{o}, 60^{o}, 120^{o} for isotropic Scotchply. The thickness of the laminates was varied and was approximately .010 per ply. See the section on Fabrication of Test Specimens for the detail process of fabrications.

Three each test specimens of the various combinations of materials and thicknesses were fabricated and tested. Each specimen was mounted in a Baldwin test machine, and a gradually increasing load was applied until the joint failed. The test setup is shown in Figure 60. Typical failures of the specimens are shown in Figures 61, 62, and 63.

Examination of the data collected on riveted joints indicates no clear-cut advantage in joint strength for one material over the others. The scatter of data between specimens of one material was often greater than the difference in results of different materials. The apparent inconsistencies in some of the results leads to the suspicion that riveting procedures may have significant influence on the results. It is suspected that rivet expansion during driving may have attributed to premature failure of some specimens. It is recommended that this factor be investigated in any future test program.

The results of the test of bolted joints indicate a definite strength advantage for the Scotchply 1002 with isotropic fiber orientation. In unsymmetrical joints using hex-head bolts, this material exhibited approximately twice the load-carrying ability of the weakest material, which was Scotchply 1002 cross-plied. The 181 cloth with epoxy resin and the 181 cloth with polyester resin exhibited approximately equal strength, with the values falling midway between the strongest and the weakest materials. In unsymmetrical joints using flush bolts, the isotropic material again showed significant advantage. The other three materials exhibited approximately equal strength with the values about two-thirds of the values for the isotropic material. Tests of symmetrical joints with hex-head bolts again showed a big advantage for the isotropic material, although the tests did not include enough specimens to establish any quantitative comparison. When metal inserts were used in the plastics, the strength was approximately the same for all the materials.

In general, the tests provide an approximation of the strength range which may be expected of a specific fastener in a specific material; it also provides an indication of the relative strength of bolted joints in different materials.

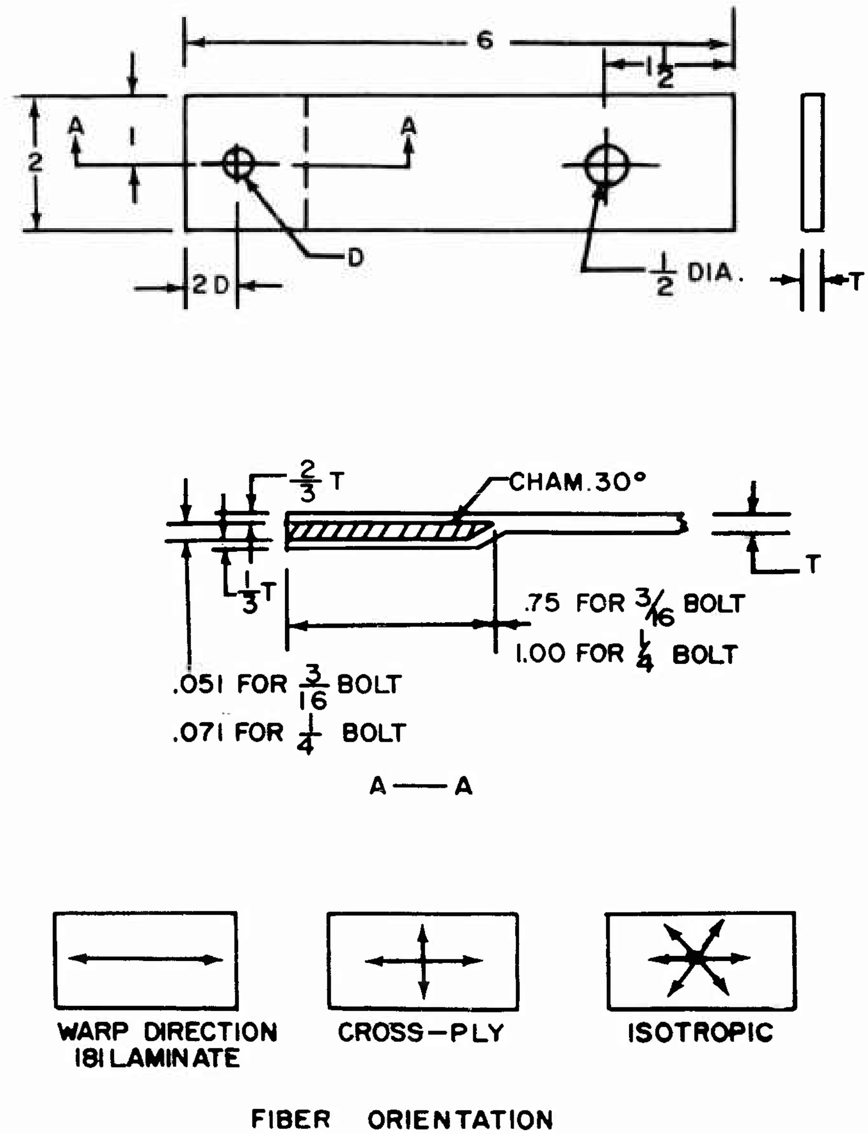


FIGURE 59. BOLTED ATTACHMENT TEST SPECIMENS
WITH MOLDED-IN METAL INSERTS

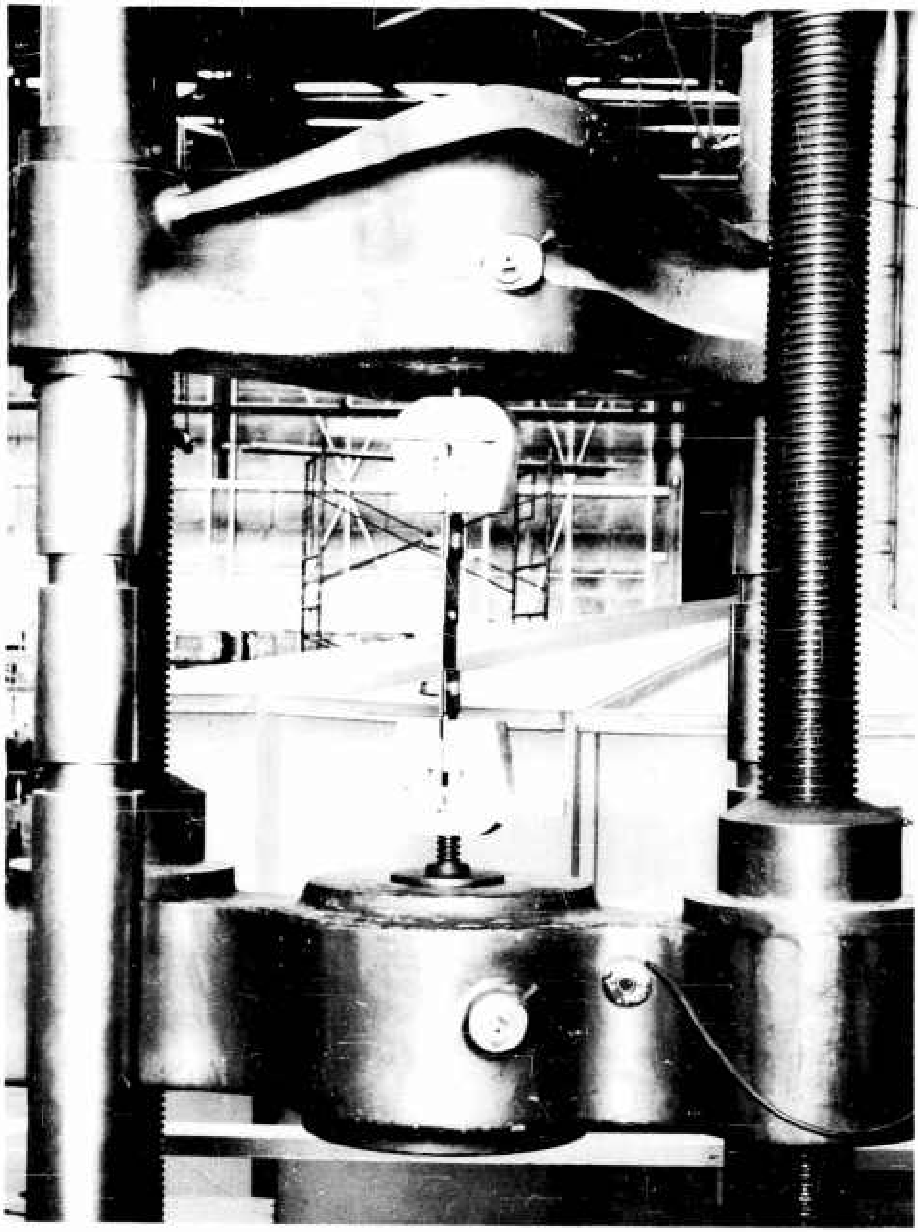


FIGURE 60. TEST SETUP FOR FASTENER TESTS

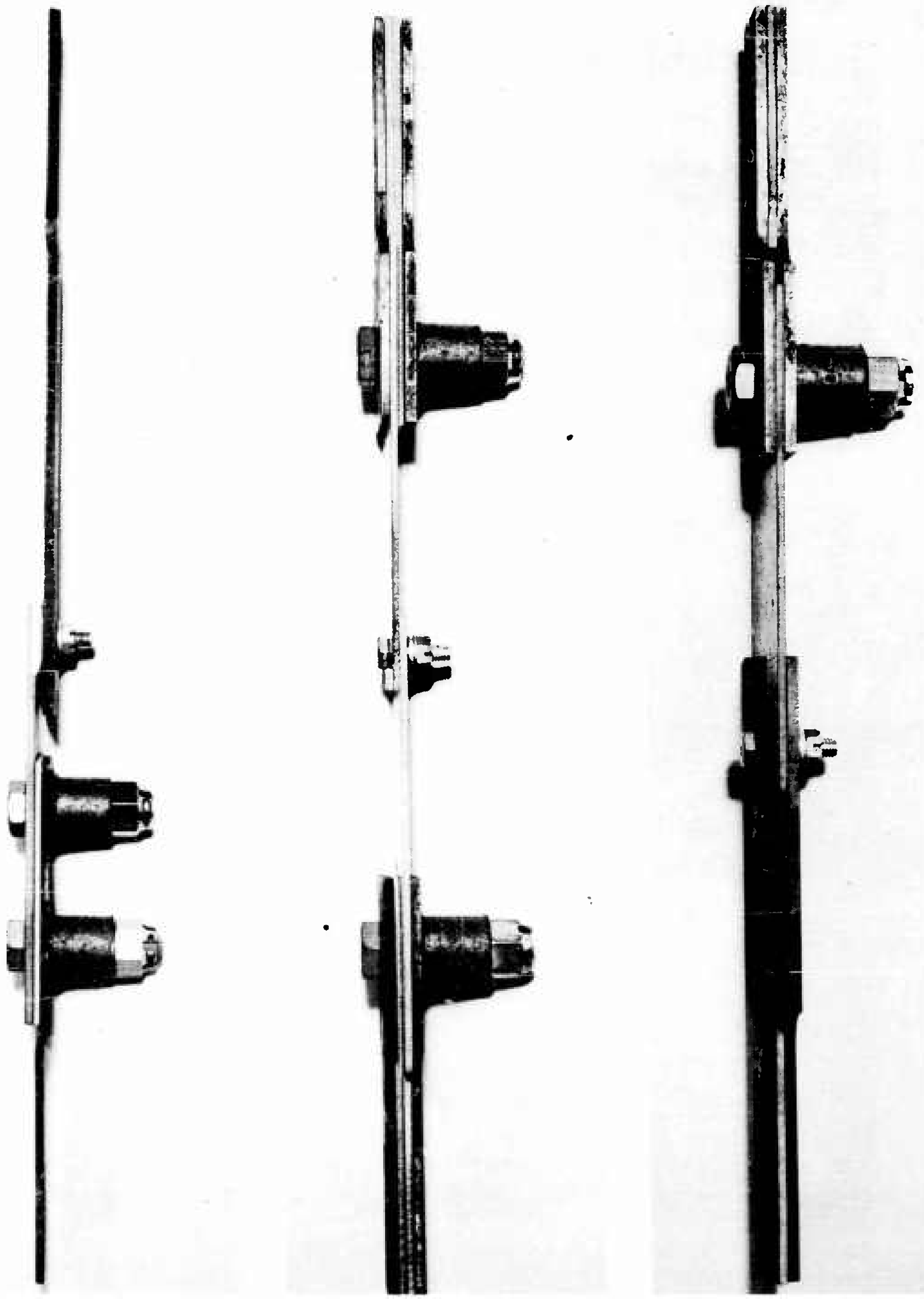


FIGURE 61. TEST ARRANGEMENT FOR FASTENER TESTS

TABLE 25

TEST RESULTS - STRENGTH OF PROTRUDING-HEAD RIVETED UNSYMMETRICAL JOINTS

Rivet Diam. (in.)	Number of Plies	Specimen Number	FAILING LOAD (lb.) (2) 181 Cloth Epoxy Resin	181 Cloth Polyester Resin	Type 1002 Scotchply Cross-Plied	Type 1002 Scotchply Isotropic
1/8	3	A	212 d	120 d	-	144 d
1/8	3	B	208 d	164 d	-	184 d
1/8	3	C	212 d	144 d	-	145 d
1/8	6	A	418 d	392 d	-	428 d
1/8	6	B	420 d	368 d	-	380 d
1/8	6	C	396 d	382 d	-	449 d
1/8	9	A	528 f	524 d	-	526 d
1/8	9	B	510 f	472 f	-	502 f
1/8	9	C	504 f	519 d	-	496 d
5/32	4	A	318 d	330 d	276 b	-
5/32	4	B	320 d	328 d	260 b	-
5/32	4	C	336 d	324 d	270 b	-
5/32	8	A	606 d	470 d	525 b	-
5/32	8	B	642 d	530 d	506 b	-
5/32	8	C	624 d	460 d	500 b	-
5/32	12	A	666 f	732 d	644 b	-
5/32	12	B	774 f	752 d	728 b	-
5/32	12	C	724 f	724 d	700 b	-
3/16	9	A	794 c	538 d	-	-
3/16	9	B	696 a	506 d	-	-
3/16	9	C	736 a	548 d	-	-
3/16	12	A	960 d	736 d	-	-
3/16	12	B	1000 a	880 d	-	-
3/16	12	C	990 c	930 d	-	-

1. Rivets were type MS 20470 AD.
2. Letter after load indicates type of failure: a) tension; b) shear tear-out; c) comb. bearing, shear and tension; d) bearing; f) rivet shear.
3. Barcol hardness readings of test specimens varied between 50 & 70.
4. Refer to Figure 59 for test specimen details.

TABLE 26
TEST RESULTS - STRENGTH OF FLUSH RIVETED UNSYMMETRICAL JOINTS

Rivet Diam. (in.)	Number of Plies	Specimen Number	FAILING LOAD (lb.) (2) 181 Cloth Epoxy Resin	181 Cloth Polyester Resin	Type 1002 Scotchply Cross-Plied	Type 1002 Scotchply Isotropic
1/8	6	A	368 b	396 e	-	406 d
1/8	6	B	366 b	404 e	-	376 d
1/8	6	C	370 b	412 e	-	384 d
1/8	9	A	498 d	480 d	-	544 f
1/8	9	B	486 d	572 d	-	556 f
1/8	9	C	510 d	492 d	-	542 d
1/8	12	A	502 f	524 f	-	480 f
1/8	12	B	514 f	548 f	-	492 f
1/8	12	C	524 f	532 f	-	474 f
5/32	8	A	586 d	520 d	562 d	-
5/32	8	B	612 d	564 d	564 d	-
5/32	8	C	584 d	552 d	560 d	-
5/32	10	A	668 d	610 d	656 d	-
5/32	10	B	580 d	672 d	634 d	-
5/32	10	C	620 d	662 d	604 d	-
5/32	12	A	766 d	700 d	680 d	-
5/32	12	B	867 d	742 d	660 d	-
5/32	12	C	806 f	764 d	656 d	-
3/16	12	A	950 d	826 d	-	-
3/16	12	B	934 d	816 d	-	-
3/16	12	C	966 d	864 d	-	-
3/16	16	A	990 f	978 d	-	-
3/16	16	B	1126 d	1078 d	-	-
3/16	16	C	1116 d	1104 d	-	-

1. Rivets were type MS 20426 AD.
2. Letter after load indicates type of failure: b) shear tear-out; d) bearing; e) comb. bearing and shear tear-out; f) rivet shear.
3. Barcol hardness readings of test specimens varied between 50 and 74.
4. Refer to Figure 59 for test specimen details

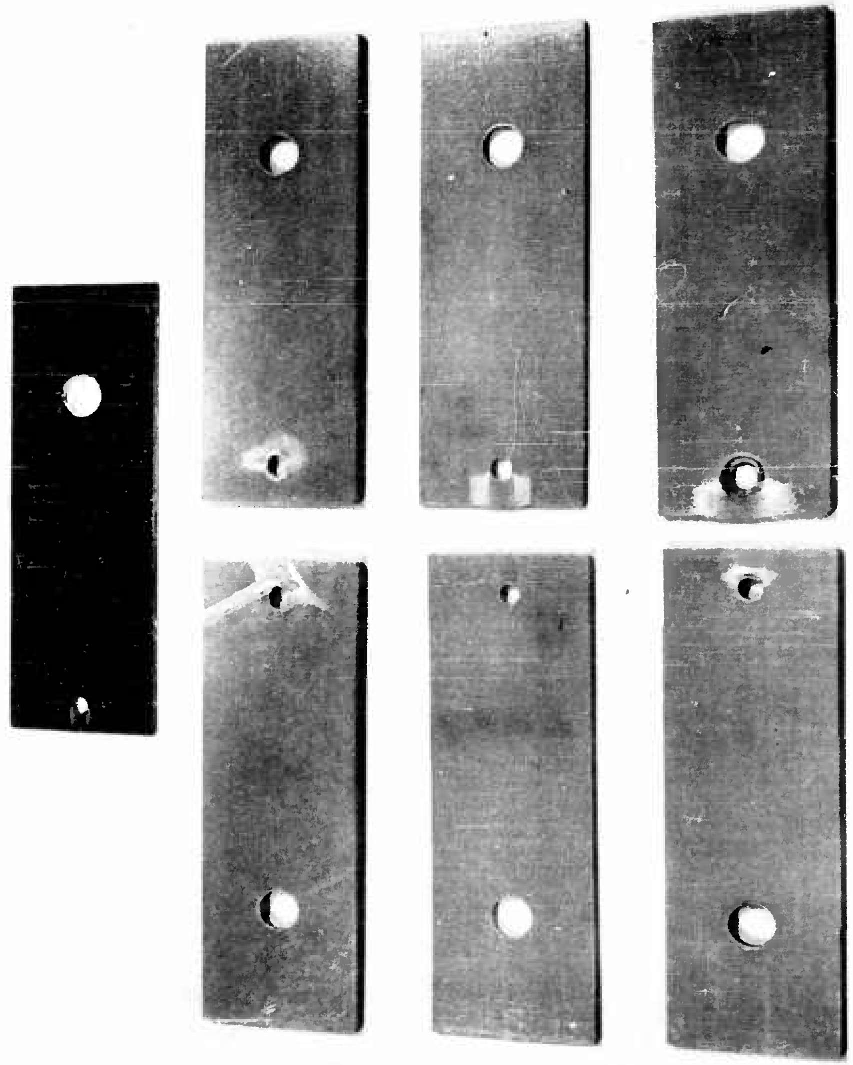


FIGURE 62. TYPICAL SHEAR TEAR-OUT FAILURES EXHIBITED IN FASTENER TESTS

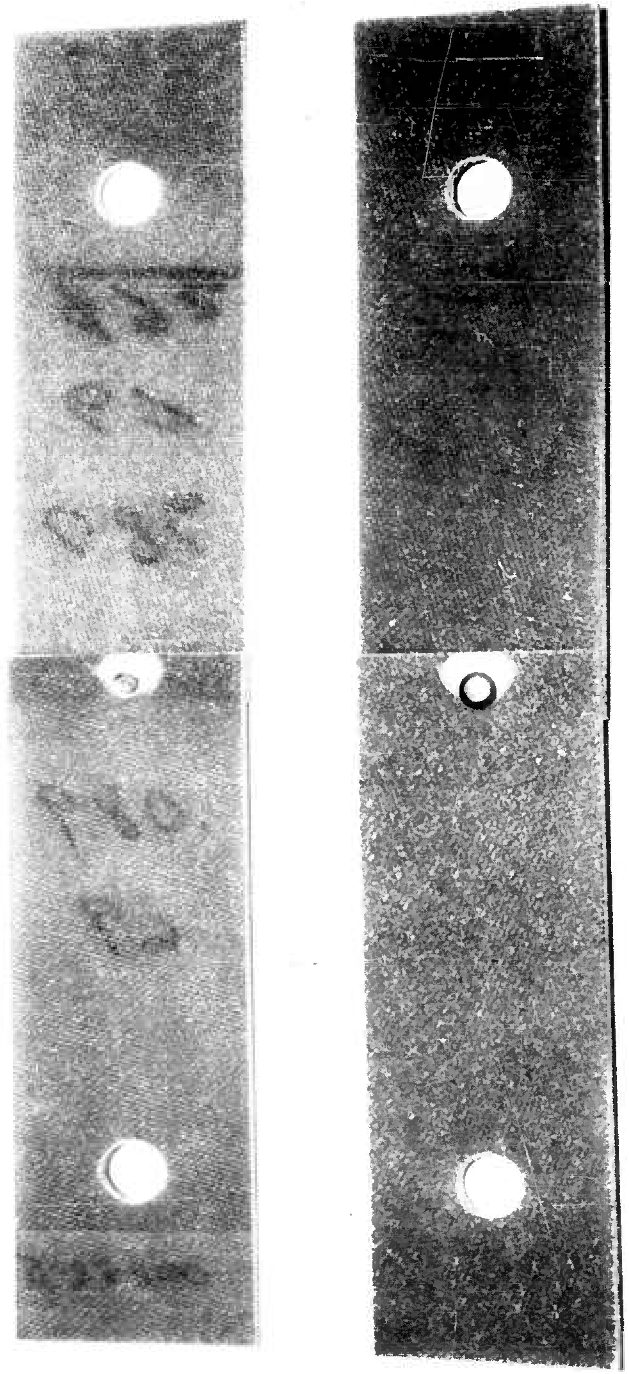


FIGURE 63. TYPICAL BEARING FAILURES EXHIBITED IN FASTENER TESTS

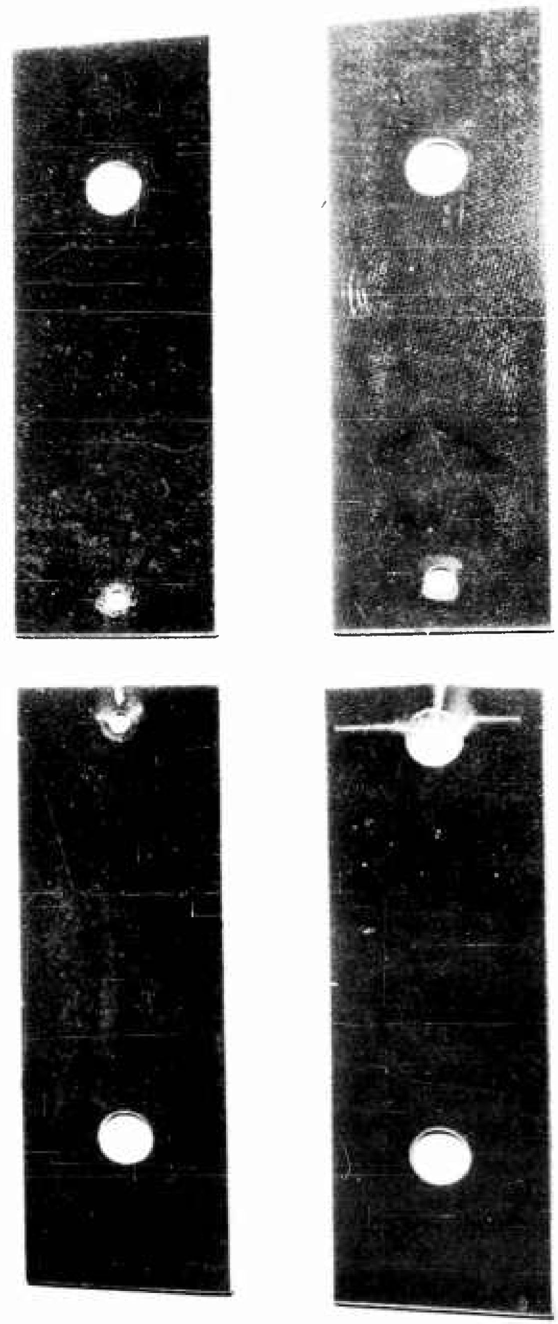


FIGURE 64. TYPICAL TENSILE FAILURES EXHIBITED IN FASTENER TESTS

TABLE 27

TEST RESULTS - STRENGTH OF HEX-HEAD BOLTED SYMMETRICAL JOINTS

Bolt Diam. (In.)	Number of Plies	Specimen Number	TYPE 181 CLOTH Failing Load (lb.)(2) Epoxy Resin	Polyester Resin	TYPE 1002 SCOTCHPLY Failing Load (lb.)(2) Cross-Plied	Isotropic
3/16	8	A	700 b	670	-	-
3/16	8	B	752 b	-	-	-
3/16	8	C	720 a & b	-	-	-
3/16	12	A	1278 b	1000	-	-
3/16	12	B	1112 b	-	-	-
3/16	12	C	1116 b	-	-	-
1/4	10	A	1190 e	1120	-	-
1/4	10	B	1260 e	1120	-	-
1/4	10	C	1328 b	-	-	-
1/4	12	A	-	1350	802 e	2170 c
1/4	12	B	-	-	746 e	2000 b
1/4	12	C	-	-	716 e	1960 d
1/4	16	A	2032 b	1810	-	-
1/4	16	B	2080 b	-	-	-
1/4	16	C	1962 b	-	-	-
1/4	18	A	-	-	1830 b	3500 a
1/4	18	B	-	-	2090 b	3652 a
1/4	18	C	-	-	2000 b	3730 a

1. Bolts were type AN-3 and AN-4.
2. Letter after load indicates type of failure: a) tension; b) shear tear-out; c) comb. bearing shear and tension; d) bearing; e) comb. bearing and shear.
3. Barcol hardness readings of test specimens varied between 58 & 71.
4. Data for polyester unpregnated specimens extracted from curves developed in a previous program (Ref. Hayes Report ER-431).
5. Refer to Figure 59 for test specimen details.

TABLE 28

TEST RESULTS - STRENGTH OF HEX-HEAD BOLTED UNSYMMETRICAL JOINTS

Rivet Diam. (in.)	Number of Plies	Specimen Number	FAILING LOAD (lb.) (2) 181 Cloth Epoxy Resin	181 Cloth Polyester Resin	Type 1002 Scotchply Cross-Plied	Type 1002 Scotchply Isotropic
3/16	6	A	640 d	618	530 b	844 e
3/16	6	B	659 d	-	525 b	790 e
3/16	6	C	660 d	-	465 b	808 d
3/16	12	A	1354 c	-	1024 b	1630 c
3/16	12	B	1318 a	-	1064 b	1340 e
3/16	12	C	1416 a	-	1040 b	1630 a
3/16	14	A	-	-	1376 b	-
3/16	14	B	-	-	1302 b	-
3/16	14	C	-	-	1346 b	-
3/16	15	A	-	-	-	2750 c
3/16	15	B	-	-	-	2800 c
3/16	15	C	-	-	-	2720 d
1/4	8	A	-	1025	870 b	-
1/4	8	B	-	-	792 b	-
1/4	8	C	-	-	790 b	-
1/4	9	A	1232 a	1150	-	1556 e
1/4	9	B	1148 a	-	-	1510 e
1/4	9	C	1234 a	-	-	1550 d
1/4	10	A	-	1280	1010 b	-
1/4	10	B	-	-	1040 e	-
1/4	10	C	-	-	1040 b	-
1/4	12	A	1660 d	1530	-	2230 b
1/4	12	B	1830 d	-	-	2110 b
1/4	12	C	1545 d	-	-	2280 e
1/4	14	A	-	-	1392 b	-
1/4	14	B	-	-	1462 b	-
1/4	14	C	-	-	1404 b	-
1/4	15	A	-	-	-	2690 d
1/4	15	B	-	-	-	5080 d
1/4	15	C	-	-	-	2860 d

1. Bolts were type AN-3 and AN-4.
2. Letter after load indicates type of failure: a) tension; b) shear tear-out; c) comb. bearing shear and tension; d) bearing; e) comb. bearing and shear.
3. Barcol hardness readings of test specimens varied between 55 & 75.
4. Data for polyester impregnated specimens extracted from curves developed in a previous program (Ref. Hayes Report ER-431).
5. Refer to Figure 59 for test specimen details.

TABLE 29
TEST RESULTS
STRENGTH OF HEX-HEAD BOLTED UNSYMMETRICAL JOINTS WITH ALUMINUM INSERTS

Bolt Diam. (in.)	Number of Plies	Insert Thick-ness (in.)	Specimen Number	FAILING LOAD (lb.) (2) 181 Cloth Epoxy Resin	Type 1002 Scotchply Cross-Plied	Type 1002 Scotchply Isotropic
3/16	8	.051	A	-	1900 b	-
3/16	8	.051	B	-	1840 b	-
3/16	8	.051	C	-	1810 b	-
3/16	9	.051	A	1952 b	-	-
3/16	9	.051	B	1886 b	-	-
3/16	9	.051	C	1942 b	-	-
3/16	15	.051	A	-	-	2600 b
3/16	15	.051	B	-	-	2690 b
3/16	15	.051	C	-	-	2480 b
1/4	9	.051	A	-	-	1940 b
1/4	9	.051	B	-	-	2020 b
1/4	9	.051	C	-	-	1930 b
1/4	12	.071	A	3455 b	3860 b	3760 b
1/4	12	.071	B	3460 b	3470 b	3700 b
1/4	12	.071	C	3450 b	3750 b	3700 b
1/4	15	.071	A	-	-	4100 b
1/4	15	.071	B	-	-	4120 b
1/4	15	.071	C	-	-	4260 b

1. Bolts were type AN-3 and AN-4.
2. Letter after load indicates type of failure: b) shear tear-out.
3. Barcol hardness readings of test specimens varied between 60 & 75.
4. Refer to Figure 59 for test specimen details.
5. Inserts were made of 2024 T-4 aluminum.

TABLE 30
TEST RESULTS - STRENGTH OF FLUSH BOLTED UNSYMMETRICAL JOINTS

Rivet Diam. (In.)	Number of Plies	Specimen Number	FAILING LOAD (lb.) (2) 181 Cloth Epoxy Resin	181 Cloth Polyester Resin	Type 1002 Scotchply Cross-Plied	Type 1002 Scotchply Isotropic
3/16	9	A	640 d	720	-	-
3/16	9	B	630 d	-	-	-
3/16	9	C	630 d	-	-	-
3/16	12	A	980 c	970	906 b	1256 b
3/16	12	B	1062 c	-	910 b	1228 b
3/16	12	C	1020 a	-	900 b	1214 d
3/16	15	A	1366 d	-	-	1955 d
3/16	15	B	1346 d	-	-	1730 d
3/16	15	C	1394 d	-	-	1904 d
3/16	16	A	-	-	1426 c	-
3/16	16	B	-	-	1376 e	-
3/16	16	C	-	-	1370 c	-
1/4	10	A	-	960	1025 b	-
1/4	10	B	-	-	920 b	-
1/4	10	C	-	-	940 b	-
1/4	12	A	1220 a	1150	-	1794 d
1/4	12	B	1160 a	-	-	1820 d
1/4	12	C	1320 a	-	-	1770 d
1/4	15	A	1600 d	1440	-	2400 d
1/4	15	B	1520 d	-	-	2290 d
1/4	15	C	1580 d	-	-	-
1/4	16	A	-	1535	1600 e	-
1/4	16	B	-	-	1545 e	-
1/4	16	C	-	-	1700 e	-
1/4	18	A	-	1730	-	2960 d
1/4	18	B	-	-	-	2510 d
1/4	18	C	-	-	-	2810 d

1. Bolts were type AN 509.
2. Letter after load indicates type of failure: a) tension; b) shear tear-out; c) comb. bearing shear and tension; d) bearing; e) comb. bearing and shear.
3. Barcol hardness readings of test specimens varied between 53 & 73.
4. Data for polyester impregnated specimens extracted from curves developed in a previous program (Ref. Hayes Report ER-431).
5. Refer to Figure 59 for test specimen details.

TABLE 31
TEST RESULTS
STRENGTH OF FLUSH BOLTED UNSYMMETRICAL JOINTS WITH ALUMINUM INSERTS

Bolt Diam. (in.)	Number of Plies	Specimen Number	FAILING LOAD (lb.) (2) 181 Cloth Epoxy Resin	Type 1002 Scotchply Cross-Plied	Type 1002 Scotchply Isotropic
3/16	8	A	-	2265 b	-
3/16	8	B	-	2340 b	-
3/16	8	C	-	2440 b	-
3/16	9	A	2170 g	-	2200 b
3/16	9	B	2140 g	-	2300 b
3/16	9	C	2150 g	-	2270 b
3/16	15	A	-	-	2580 f
3/16	15	B	-	-	2560 f
3/16	15	C	-	-	2520 f
1/4	12	A	3300 c	3310 b	3710 b
1/4	12	B	3490 c	3300 b	3600 b
1/4	12	C	3380 c	3470 b	3600 b
1/4	15	A	-	-	3540 b
1/4	15	B	-	-	3720 b
1/4	15	C	-	-	3450 b

1. Bolts were type AN-509.
2. Letter after load indicates type of failure: b) shear tear-out; c) comb. bearing, shear and tension; f) bolt shear; g) tension and delamination.
3. Barcol hardness readings of test specimens varied between 60 & 72.
4. Refer to Figure 59 for test specimen details.
5. All inserts were 2024 T-4 aluminum, .071 in. thick.

Adhesive Bonding

In the use of reinforced plastics for primary structure in Army aircraft, it will be necessary to make many attachments, splices, and joints. One method of joining structural parts is by adhesive bonding. Adhesive bonding of aircraft structure has been used extensively from the early days of World War II (British Mosquito Aircraft) to the present usage in such modern aircraft as the B-58 supersonic bomber. The adhesive bonding of reinforced plastics has been successful in nonstructural aircraft applications. The adhesive manufacturers have little or no data on the properties of an adhesive joint in reinforced plastic. Some adhesive manufacturers claim that "the adhesive bond is always stronger than the plastic itself".

Adhesives should find satisfactory use in the bonding of primary structures fabricated of reinforced plastic because of the established advantages (weight reduction and better fatigue life) of metal bonding and the natural compatibility of the plastic resin and the adhesive resin.

About five years ago, there became available a new type of adhesive, called epoxy, with properties far superior to the older types of adhesives. Epoxies are two-part materials which do not require solvent evaporation during curing. Epoxy resins cure by chemical reaction of the base resin with a catalyst. Their shrinkage during cure is very slight, and they are compatible with the resin systems used for reinforced plastics. Epoxies have good wetting ability, which enables them to penetrate small pits; and their low viscosity before cure allows them to flow easily between the surfaces to be bonded. Other types of resin systems which have found use in structural bonding are the epoxy-phenolics, nitrile-phenolics, and vinyl-phenolics.

It has been found that the ultimate breaking strength of an adhesive is influenced by the type of material being bonded. For example, with the same adhesive it will be found that a metal-to-metal joint is generally stronger than a joint in which one or both of the materials are a reinforced plastic. Therefore, to evaluate an adhesive joint properly, a full history of base material, surface preparation, cure cycle, etc., must be known.

There is a choice of the types of adhesive systems which might be used on structural parts. One variation involves the physical form of the adhesive. It may be either in a liquid form or in a film form. Another variation is the cure system for the adhesive. Some adhesives require heat to produce the cure while others will cure at room temperature. In addition, the adhesive can be adjusted in viscosity, working life, color, etc. Film-type adhesives usually require heat and pressure to effect the proper cure. Liquid adhesives can be cured either at room temperature or at an elevated temperature.

The strength of adhesives is usually measured by a simple lap shear joint using a ½-inch or a 1-inch overlap of sheet specimens 1 inch wide and 4 inches long with a thickness of .064 inch, .100 inch, etc. The specimens are bonded together, cured, and then pulled in tension to failure. The shear stress is calculated by dividing the failing load by the bonded area. Military specifications for structural adhesives provide for a shear test to be run for qualification and process control purposes. All of these shear tests of adhesives are designated to be run on metal-to-metal specimens. The shear strength of structural adhesives at room temperature on metal-to-metal specimens is about 2500 p.s.i. maximum.

As previously mentioned, no data were found to be available from adhesive manufacturers on the shear strength of adhesive bonded reinforced plastic joints. The military specifications do not report minimum shear strength for reinforced plastic adhesive bonded joints. Therefore, some tests were run to obtain the desired data.

Two adhesive manufacturers, Bloomingdale Rubber Company and Minnesota Mining and Manufacturing Company, offered their services to bond-test specimens and to conduct shear tests of the bonded joints. The test specimens were 1 x 5 x .100 inches. It was requested that the specimens be bonded with a 1-inch overlap, cured as required, and then tested. Two types of adhesive were requested to be used: one a room temperature curing liquid and the other a heat and pressure curing film.

The test results along with pertinent data concerning the test specimens are presented in Tables 32 and 33. These data indicate that relatively high shear strengths can be obtained in reinforced plastics. However, the data are very limited, and it is evident that more work needs to be done to obtain statistically reliable data. The variables which need to be explored are: resin system, adhesive thickness, surface preparation, overlap length, cure pressure, cure temperature, wetting, fit-up tolerances, damping, fatigue life, strength at various temperatures, etc.

TABLE 32
SHEAR FAILING STRESSES
MINNESOTA MINING & MFG. CO. ADHESIVES FOR REINFORCED PLASTIC

Laminate		Adhesive			Shear Stress (p.s.i.)		
Reinforcement	Resin	Type	Number	Cure	-67°F	Room	+250°F
181 Cloth	Epoxy	2 Part	EC 2216	Contact	-	1850	220
"	Polyester	Epoxy	B/A	Pressure	-	2260	340
Scotchply 1002	Epoxy			Room Temp.	2000	-	370
181 Cloth	Epoxy	Unsupport-	AF-111	10 psi Press.	-	-	180
"	Polyester	Film		1 hr.@250°F	2800	-	410
Scotchply 1002	Epoxy				-	2330	370
181 Cloth	Epoxy		AF-40	100 psi Press.	3120	-	490
"	Polyester	"		1 hr.@300°F	-	2250	1170
Scotchply 1002	Epoxy				-	1960	980
Reinf. Plastic*		2 Part	EC1648B/A	Contact	1710	1780	-
"		Epoxy	EC1838B/A	Room Temp.	1860	2160	-

*Not Identified - Data supplied by 3M - Specimens not supplied by Hayes

TABLE 33

SHEAR FAILING STRESSES

BLOOMINGDALE RUBBER CO. ADHESIVES FOR REINFORCED PLASTIC

Speciman No.	Laminate		Adhesive			Shear Stress (p.s.i.)
	Reinforcement	Resin	Type	Number	Cure	
90-227	181 Glass Cloth	Epoxy	2 Part Epoxy Liquid	BR-90 with 8 PHA "A" Accel.	Contact Pressure 7 Days Rm. Temp.	851 850 816
90-228	181 Glass Cloth	Polyester	"	"	"	800 662 825
90-229	Scotchply 1002 Crossply	Epoxy	"	"	"	977 907
62-4100	181 Glass Cloth	Epoxy	Unsupport-ed Film	FM-1000	30 min. to 350°F, 60 min. at 350°F, 25 psi	1795 1700
62-4101	181 Glass Cloth	Polyester	"	"	"	2675 2024 1660
62-4102	Scotchply 1002 Crossply	Epoxy	"	"	"	3450 3470 3010

All laminates were approximately .10 inch thick

Filament Winding

Filament winding applications in the past have been largely confined to pressure vessels and related types of components. It is believed that the process has potential applications for certain structural components such as body, wing, empennage and control surface structure. Fuel tanks, tubes, and ducts are also potential applications for filament winding. An aircraft company in Sweden has successfully tested a fixed landing gear strut of fiberglass reinforced plastic fabricated by a winding process.

In order to realize the full potential advantages of the higher strength and rigidity of filament-wound structures, certain basic design principles should be considered:

1. Design initially for filament winding.
2. Use a shape that can be wound under tension.
3. Close tolerances can be held with filament winding.
4. Cost per part can be low.

In this process the part is wound on a mandrel whose exterior represents the interior configuration of the part. The mandrel is removed after the part is formed and cured. Reverse curvatures are extremely difficult to wind. Certain types can be produced by specialized techniques, but strength characteristics will not be optimum. Parts consisting of surfaces of revolution are ideal for winding.

The H-23 tail boom investigated in this program is an excellent example of structure that can be feasibly and probably advantageously filament wound.

A rather new material that is extremely interesting is a filament-orientated preimpregnated material. The purpose of this material is to combine the preorientation of filaments inherent in filament winding with the shape flexibility inherent in molding flat preimpregnated reinforced plastics.

The material was developed by Hercules Powder Company and is made by winding impregnated glass roving on a cylindrical mandrel in a pre-determined helix. The cylindrical structure thus produced is then slit axially, flattened and molded, or the material can be B-staged. The material can be molded by bag or matched metal techniques.

Winding with a 45-degree helix results in a preimpregnated material with fibers orientated at 90 degrees to each other. Different helix angles can be used to provide various degrees of directionability in strength. The material is somewhat similar to Minnesota Mining and Manufacturing Company's Scotchply. However, in the filament-wound preimpregnated material, there is a degree of "over and under" interweaving between plies to form a very loose basket-type structure.

This permits a substantial amount of "wash" of fiber in deep draw molding, allowing the material to conform smoothly to deep hemispherical drawn and compound curvatures.

Although these materials are still developmental, they appear to be highly promising for a variety of molding applications. Components where certain aircraft body complex shapes are encountered, yet maximum strength and rigidity are required, appear to be applications where they could be used advantageously.

Many highly successful parts have been fabricated by filament winding. However, there are virtually no specific design data generally available. Most available strength data indicate only general orders of magnitude obtainable, and cannot usually be related directly to a specific structure.

As stated in the introduction to this section, Hercules Powder Company was contacted to provide certain data on bending, axial compression, shear, torsion, and natural frequency of filament-wound, epoxy resin-bonded fiberglass structures. Following is a summary of these data. The term "Spiralloy" relating to the filament-wound structure is a registered trade name used by Hercules Powder Company. Limited information on their filament-oriented preimpregnated Spiralloy mat is also included.

Summary

Design properties for bending, flexure modulus of elasticity, axial compression and shear have been reviewed and collected in this report. Most of the test data apply to 15° helical windings. These data have been extended by the use of the netting analysis to calculate properties for a range of other angles. In line with this, a conversion chart is presented; this chart will enable the designer to convert tube designs of any helix angle to an equivalent 15° tube of equal strength in order to predict its resistance to bending or compressive buckling failure.

Flexure modulus of elasticity data have been used to plot curves of "frequency constants" for various helix angles and percentages of helicals. Using the constants, it becomes a simple matter to determine the natural frequency of thin-walled Spiralloy tubes.

Torsion tests have been conducted to study the correlation between theoretical relationships and test data in analyzing the strength of various helix angles in pure torsion, the effect of the D/t ratio, and the determination of the modulus of rigidity.

The use of filament-wound mat, as applied to the design of sandwich structure walls, appears to be promising. The properties of Spiralloy mat are reported, but are based upon very limited data. Table 38 shows selected substantiating test data for strength properties of Spiralloy components presented in this report.

Bending Buckling

In Figure 65, two curves are shown for ultimate bending stress in Spiralloy thin-walled tubes. These are in the low D/t range and are plotted as a function of the percentage of 15^{o} helical layers. In all cases, circular (90^{o}) windings are used to supplement the helical winding. A minimum of approximately 20 percent circular winding is required, especially with low helix angles, to consolidate the structure and to insure a high glass density.

Figure 65 shows the bending buckling strength as a function of D/t for various percent helicals in the low D/t range, and is the basis for Figure 66.

Figure 67 is a chart for converting thin-walled tubes of other helix angles to equivalent 15^{o} tubes so that the 15^{o} curves may be utilized for other helix angles as well. In using this chart, use $N_{15} = R \times N_{\alpha}$ where N_{15} = % of 15^{o} helicals N_{α} = % of helicals at angle α.
The chart is based on the netting analysis relationship

$$\sigma_{\alpha} = \frac{Mc}{I} \quad \frac{100}{N_{\alpha} \cos^2\alpha}$$

where

σ_{α} is filament stress due to bending
M is bending moment
c is cylinder radius
I is moment of inertia about the central axis
N_{α} is % of helicals

To convert from α_1 to α_2 with the same percent helicals in both systems, the relationship will be

$$N_{\alpha_1} = N_{\alpha_2}$$

$$\sigma_{\alpha_1} = \sigma_{\alpha_2}$$

$$\frac{\sigma_1}{\cos^2 \alpha_1} = \frac{\sigma_2}{\cos^2\alpha_2}$$

$$\frac{\sigma_2}{\sigma_1} = \frac{\cos^2 \alpha_2}{\cos^2 \alpha_1}$$

To maintain the same bending strength, the percent of helicals in the new system must be changed by the factor

$$\frac{\cos^2 \alpha_2}{\cos^2 \alpha_1}$$

In this case, $\alpha_1 = 15^{o}$ and $\frac{\cos^2 \alpha_2}{\cos^2 15^{o}} = p$

The stress values in the table that follows were taken from Figure 67, utilizing the conversion chart.

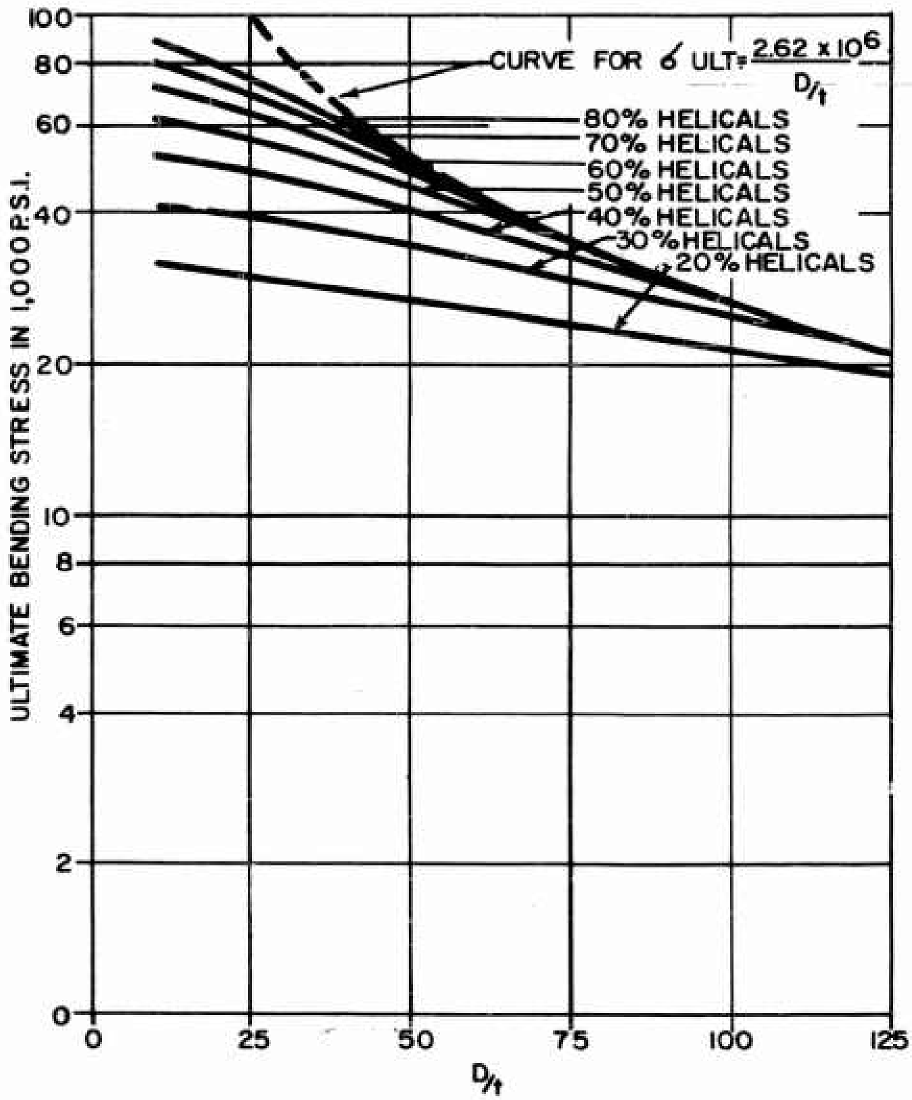


FIGURE 66. BENDING BUCKLING STRENGTH OF 15° SPIRALLOY TUBES VS. DIAMETER-THICKNESS RATIO

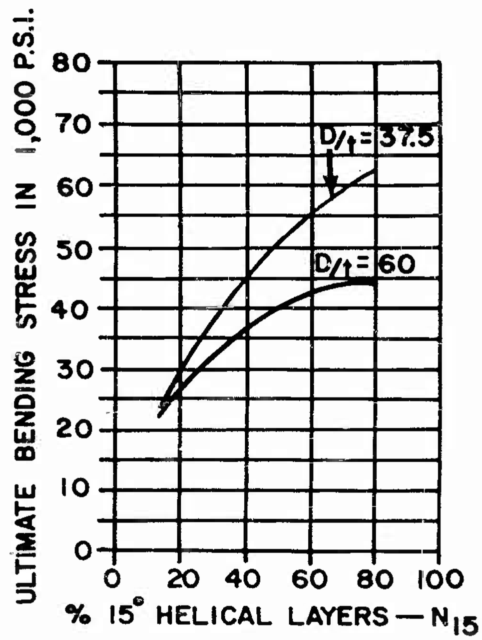


FIGURE 65. BENDING BUCKLING STRENGTH OF 15° SPIRALLOY TUBES

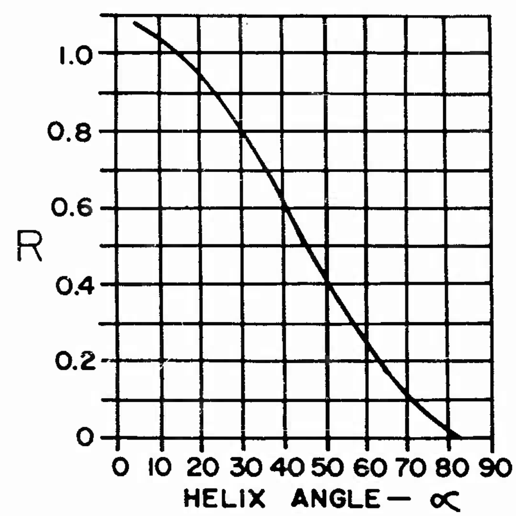


FIGURE 67. CONVERSION CHART FOR CONVERSION OF ANY SPIRALLOY TUBE USED IN FLEXURE OR AXIAL COMPRESSION TO A 15° TUBE OF EQUIVALENT STRENGTH

TABLE 34

MAXIMUM BENDING STRESSES FOR THIN-WALLED SPIRALLOY TUBES FOR D/t = 60

	HELIX ANGLE							
	25°		35°		45°		55°	
N (%)	N_{15} (%)	Ult. Stress (p.s.i.)	N_{15} (%)	Ult. Stress (p.s.i.)	N_{15} (%)	Ult. Stress (p.s.i.)	N_{15} (%)	Ult. Stress (p.s.i.)
20	17.6	24,800	14.4	22,500	-	-	-	-
30	26.4	30,500	21.6	27,500	16.1	23,800	-	-
40	35.2	35,000	28.7	31,500	21.4	27,400	14.1	22,200
50	44.0	37,800	35.9	35,200	26.8	30,600	17.6	24,800
60	52.8	41,000	43.1	38,000	32.2	33,500	21.2	27,200
70	61.6	42,800	50.2	40,500	37.6	35,800	24.7	29,500
80	70.4	43,900	57.4	42,200	42.9	38,000	28.2	31,300

TABLE 35

MAXIMUM BENDING STRESSES FOR THIN-WALLED SPIRALLOY TUBES FOR D/t = 37.5

	HELIX ANGLE							
	25°		35°		45°		55°	
N (%)	N_{15} (%)	Ult. Stress (p.s.i.)	N_{15} (%)	Ult. Stress (p.s.i.)	N_{15} (%)	Ult. Stress (p.s.i.)	N_{15} (%)	Ult. Stress (p.s.i.)
20	17.6	26,800	14.4	24,000	-	-	-	-
30	26.4	34,500	21.6	30,500	16.1	25,600	-	-
40	35.2	41,000	28.7	36,300	21.4	30,400	14.1	23,700
50	44.0	46,500	35.9	41,400	26.8	34,300	17.6	26,800
60	52.8	51,400	43.1	46,000	32.2	38,000	21.2	30,200
70	61.6	55,500	50.2	50,000	37.6	42,600	24.7	33,000
80	70.4	58,000	57.4	53,700	42.9	46,000	28.2	36,000

Natural Frequency

The graph, Figure 68, of flexure modulus of elasticity for 15° helix angle tubes has been established by test data. Using this information, values were determined for the modulus of elasticity in the direction of the helical filaments (6.2×10^6) and normal to the circular windings ($.75 \times 10^6$).

The modulus for any helix angle is the sum of the moduli of the two elements, helical layers and circular layers; therefore,

$$E_f = \frac{N_\alpha}{100} (6.2)(10^6)\cos^2\alpha + \left(1 - \frac{N_\alpha}{100}\right)(.75)(10^6)$$

where

N_α is percentage of helicals.

The remaining curves in Figure 68 were calculated by this method.

Using this basic information, curves of natural frequency constants (K) versus the helix angle for thin-walled tubes were drawn. Three percentages of helicals and two methods of beam support were considered. The general formulas (from Alcoa Structural Handbook, Copyright 1956, Page 185) are as follows:

(1) Cantilever support

$$f = \frac{3.89}{\sqrt{D}} = \frac{3.89}{\sqrt{\frac{w\ell^4}{8\ EI}}} = \frac{K_c\ R}{\ell^2}$$

where

- f is natural frequency in cycles per sec.
- D is deflection, in.
- w is distributed load, lb./in.
- l is tube length, in.
- I is moment of inertia $= \pi R^3$ t for a thin-walled tube.
- E is modulus of elasticity per Figure 68.
- K_c is "frequency constant" for cantilever support per Figure 69.
- R is mean tube radius, in.

(2) Simple support

$$f = \frac{3.55}{D}$$

$$= \frac{3.55}{\sqrt{\frac{5}{384} \frac{w \ell^4}{EI}}}$$

$$= \frac{K_s R}{\ell^2}$$

where

K_s is "frequency constant" for simple support per Figure 70.

A specific example was also worked out andpplotted in Figure 71. This is for a cantilevered Spiralloy tube with a 3-inch nominal diameter, 48 inches long. The natural frequency is plotted against the helix angle for 40, 60 and 80 percent helicals.

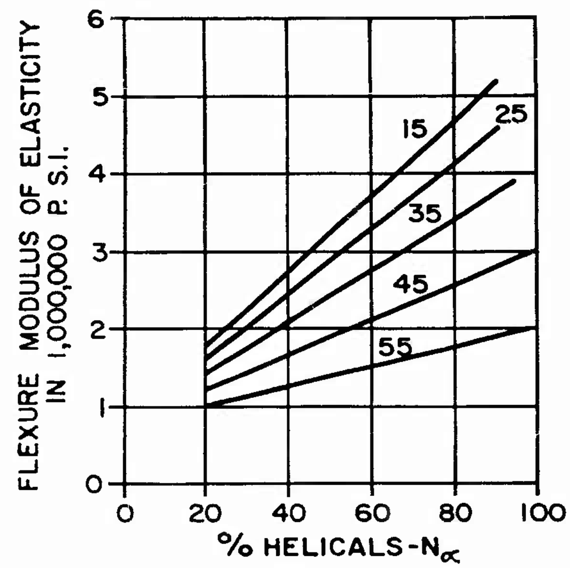


FIGURE 68. AVERAGE FLEXURE MODULUS OF ELASTICITY FOR THIN-WALLED SPIRALLOY TUBES

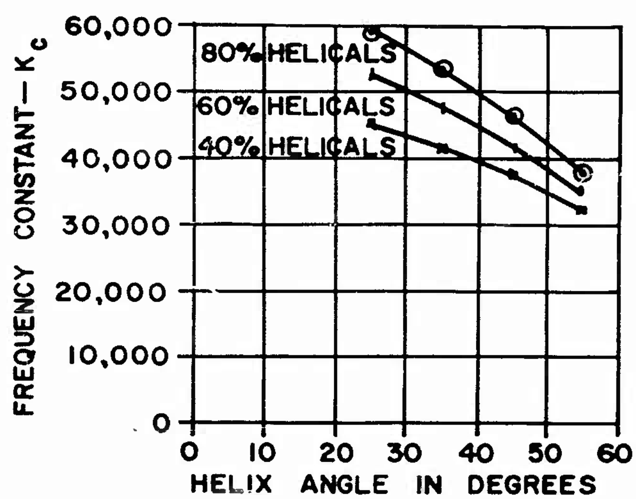


FIGURE 69. FREQUENCY CONSTANT VS. HELIX ANGLE FOR CANTILEVER THIN-WALLED TUBE

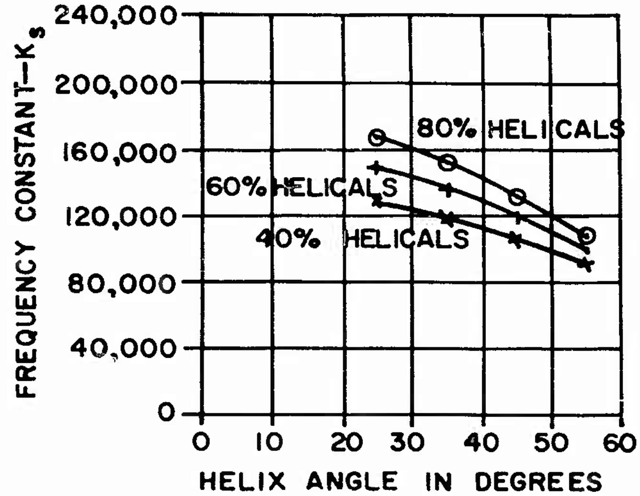


FIGURE 70. FREQUENCY CONSTANT VS. HELIX ANGLE FOR SIMPLY SUPPORTED THIN-WALLED TUBE

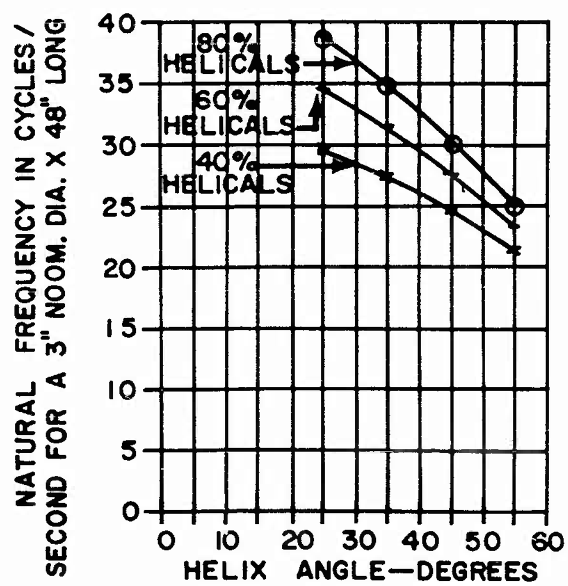


FIGURE 71. NATURAL FREQUENCY VS. HELIX ANGLE FOR A CASE OF A CANTILEVERED THIN-WALLED TUBE 3" NOMINAL DIAMETER X 48" LONG

Axial Compression

Figure 72, which follows, shows a family of curves for the axial compressive buckling strength of 15^{o} Spiralloy tubes. It should be noted that in the high D/t range, the buckling stress level is essentially unaffected by the percent of helicals, whereas in the low D/t range, the effect is pronounced.

The plotted buckling curve for values of D/t above 75 follows the relationship from Roark, "Formulas for Stress and Strain" (3rd Edition), P. 316, case M reduced by a factor in the order of .64

$$\sigma \text{ ult.} = \frac{.64}{\sqrt{3}} \frac{E}{\sqrt{1 - \mu^2}} \left(\frac{t}{r}\right)$$

where

E is composite modulus of elasticity in compression = 3.4×10^6
μ is Poisson's Ration = .28

$$= \frac{2.62 \times 10^6}{D/t}$$

As for bending, the conversion chart, Figure 67, was employed to obtain the stress values for other helix angles from Figure 72.

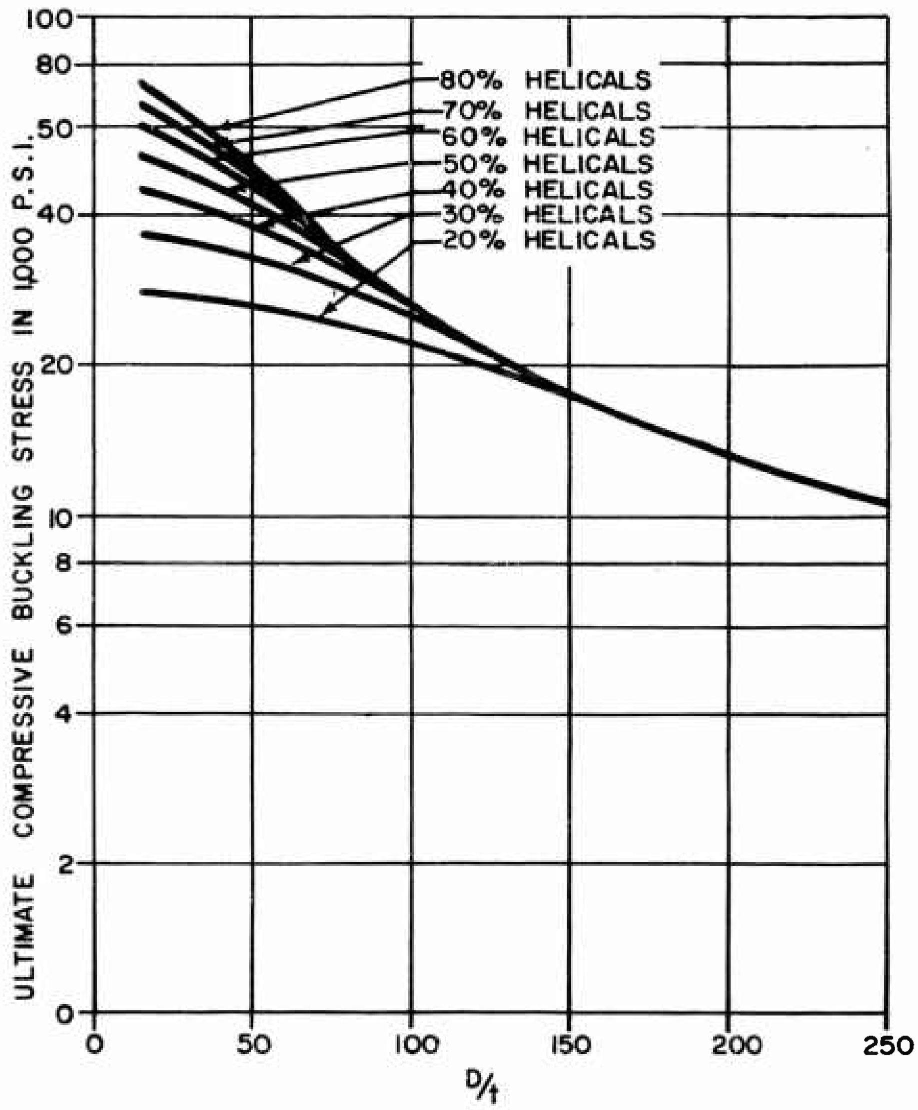


FIGURE 72. AXIAL COMPRESSIVE BUCKLING STRENGTH OF 15° SPIRALLOY TUBES VS. DIAMETER-THICKNESS RATIO

TABLE 36

MAXIMUM AXIAL COMPRESSIVE STRESSES FOR THIN WALLED SPIRALLOY TUBES

$\alpha = 25°$				$\alpha = 35°$			
N (%)	Equivalent N_{15} (%)	Ultimate Stress (p.s.i.) D/t = 60	D/t = 37.5	N (%)	Equivalent N_{15} (%)	Ultimate Stress (p.s.i.) D/t = 60	D/t = 37.5
22.7	20	25,000	27,000	27.9	20	25,500	27,000
34.1	30	31,300	34,300	41.8	30	31,300	34,300
45.4	40	35,800	41,000	55.7	40	35,800	41,000
56.7	50	38,800	46,000	69.7	50	38,800	46,000
68.1	60	41,000	50,700	83.6	60	41,000	50,700
79.4	70	43,000	55,000				
$\alpha = 45°$				$\alpha = 55°$			
37.4	20	25,500	27,000	56.6	20	25,500	27,000
56.1	30	31,300	34,300	85.0	30	31,300	34,300
74.8	40	35,800	41,000				

Interlaminar And Cross Shear

Figure 73 shows the effect of the percentage of 15^{o} helicals upon the interlaminar shear strength. Interlaminar shear strength is also dependent upon the resin and the finish of the roving. A combination of 801 finish roving and 828/CL resin was used in this series of tests, but materials in use at the present time give somewhat higher values.

Cross laminar shear is presented in two closely related forms. Figure 74 shows the shear stress in unidirectional windings as a function of the angle of the shear plane. Figure 75 shows shear stress along an axial plane as a function of the helix angle.

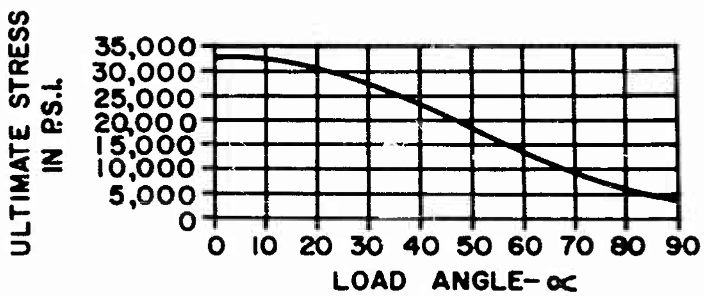


FIGURE 74. ULTIMATE CROSS SHEAR STRESS OF UNIDIRECTIONAL WINDING AT ANY ANGLE

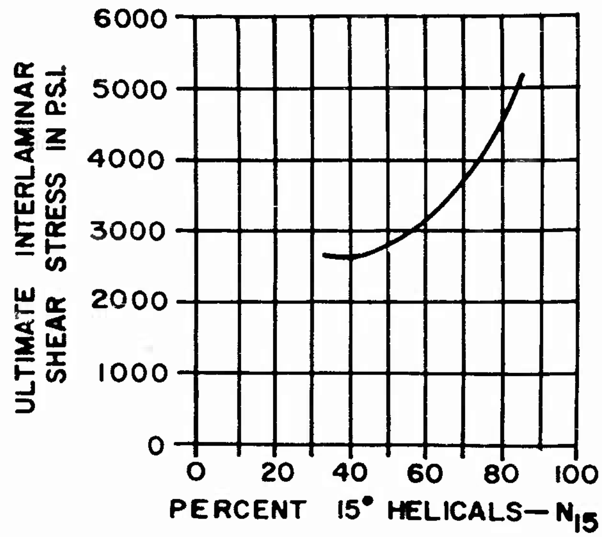


FIGURE 73. ULTIMATE INTERLAMINAR SHEAR STRENGTH VS. PERCENT 15° HELICALS

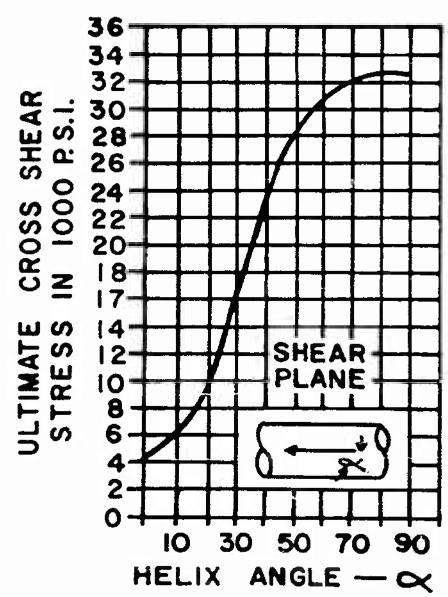


FIGURE 75. ULTIMATE CROSS SHEAR STRESS VS. WINDING ANGLE

Spiralloy Mat

Table 37, below, summarizes some limited data on the physical properties of Spiralloy mat. Spiralloy mat, since it is not woven, conforms well to a die with filaments taut; but since the possibility of filament slackness is inherent, all applications of mat might not exhibit the same properties. The values are averages of only two tests for each of three conditions.

Samples consisted of 45^{o} filament winding, freezer stored, then pressed and cured flat in a heated press to a thickness of $.125 \pm .015$ inch. The resin formulation was Shell Epon 828 and a "HET" system curing agent.

TABLE 37
AVERAGE COMPOSITE STRENGTH OF SPIRALLOY MAT

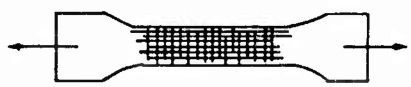
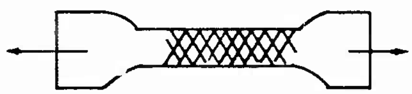
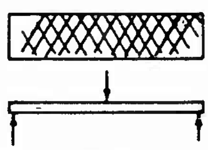
Fiber Orientation and Loading	Type of Test	Av. Ult. Composite Stress (p.s.i.)	Variation
	Tensile	93,500	20%
	Tensile	18,640	6.65%
	Bending	57,950	2.5%

TABLE 38
SELECTED SUBSTANTIATION DATA FOR STRENGTH OF SPIRALLOY

Type of Test	Nom. Dia. (in.)	D/t	α (degrees)	% α	E (p.s.i. x 10^{-6})	Ultimate Stress (p.s.i.)
Bending	4	33	15	80	-	86,000
	4	31	"	80	-	76,000
	8	50	"	79	-	49,300
	4	89	"	80	-	29,300
	4	90	"	80	-	32,100
Natural frequency	8	45	"	80	4.8	-
	8	124	"	57	3.48	-
	4	46	"	40	2.6	-
Axial compression	4-1/2	23	"	76	-	62,000
	4-1/2	24	"	79	-	69,300
	18	245	"	83	-	10,419
	18	282	"	50	-	10,250
Interlaminar shear	2.6	-	"	33	-	2,744
	2.6	-	"	83	-	4,813
	2.6	-	"	66	-	3,259
	2.6	-	"	66	-	3,746
Cross laminar shear (uni-directional winding)	-	-	"	100	-	31,825
	-	-	45	100	-	21,225
	-	-	90	100	-	3,562
Cross laminar shear (helical winding)	2.6	-	60	100	-	31,715
	2.6	-	45	100	-	24,667
	2.6	-	30	100	-	15,042
	2.6	-	90	100	-	3,562*

*Applies to $\alpha = 0^{\circ}$ for helical winding.

Torsion

Twelve sample tubes were fabricated and tested for this report. Three helix angles were wound (30, 45 and 60), each tube being composed of entirely one angle, except for a surface covering of 1/2 circular layer (less than 10 percent of total) for consolidation purposes. The resin system is Shell Epon 826 with CL curing agent, and the roving is Fiberglas E. C. G. 12 end #140 with HTS finish. This combination of resin formulation and glass roving is one in general use with a large background of useful comparative data.

Samples were tested in pure torsion with built-up ends restrained from bending or collapsing. The modulus of rigidity was determined for each sample from a plot of the torque versus angle of twist. The moduli are shown in Figure 76 superimposed on the theoretical netting analysis curve

$$G = E \sin\alpha \cos\alpha$$

where

E is composite modulus of elasticity arbitrarily taken at 3.4×10^6.

This theoretical relationship appears to be supported even though some of the points are widely spread.

Failure in each case occurred in buckling, which followed the theoretical torsional buckling relationship from Roark, "Formulas for Stress and Strain" (3rd Edition), P. 317, case O, decreased by a factor of .9.

$$\sigma_s = .9 \frac{E}{1-\mu^2} \left(\frac{t}{\ell}\right)^2 \left[4.6 + \sqrt{7.8 + 0.59 H^{3/2}}\right]$$

$$\text{and } H = \sqrt{1-\mu^2} \left(\frac{\ell^2}{tr}\right)$$

where

E is composite modulus of elasticity taken as 3.4×10^6 p.s.i.

μ is Poisson's Ratio = .28

Ultimate stress values are plotted against this curve in Figure 77. It appears quite conclusive that in this D/t range the tubes are buckling critical, and torsion test members would have to be relatively thick-walled in order that the effect of helix angle upon the ultimate shear stress may be studied.

According to the netting analysis on Page 239, if D/t is sufficiently low to eliminate buckling,

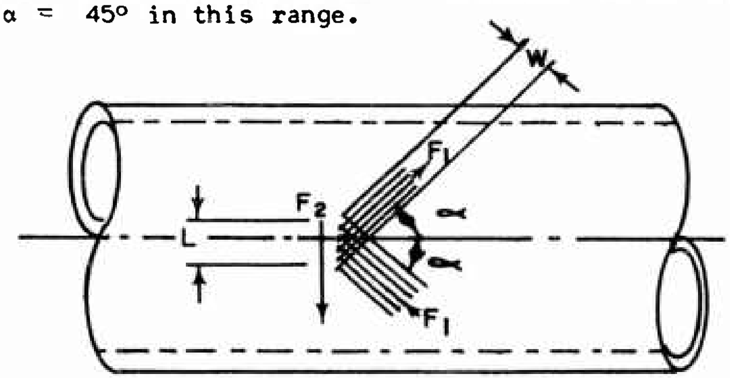
$$\sigma_s = \sigma \sin\alpha \cos\alpha$$

where

σ is unit strength parallel to the filament system = approximately 100,000 p.s.i.

$$\text{at } 45^\circ \; \sigma_s = \sigma \; (.707)^2 = 50{,}000 \text{ p.s.i.}$$

This is verified by one previous test with 45° windings in the very low D/t range. (See Table 39 and Figure 76.) The dotted portion of the curve, Figure 77, represents the anticipated deviation from the buckling curve for α = 45° in this range.



$$F_1 \approx \frac{SWt}{2}$$

$$F_2 = 2F_1 \sin\alpha = S\,W\,t \sin\alpha$$

$$S_s = \frac{F_2}{Lt}$$

$$L = \frac{W}{\cos\alpha}$$

$$S = \frac{SWt \sin\alpha}{\frac{W}{\cos\alpha}\, t} = S \sin\alpha \cos\alpha$$

where

S is unit strength of parallel filament system

S_s is unit strength of member in torsional shear

t is layer thickness (closed system of right-and left-hand helical bands)

α is helix angle

Calculation of Torsion Test Data

Ultimate Shear Stress:

$$\sigma_s = \frac{2\ T\ r_1}{\pi\ (r_1^4 - r_o^4)}$$ (Ref. Roark, "Formulas for Stress and Strain", P. 175, case 6)

where

T is load (42 in.)

Modulus of Rigidity:

$$G = \frac{2(\Delta T)\ell}{\pi(r_1^4 - r_o^4)\ (\Delta\theta)}$$ (Ref. Roark, "Formulas for Stress and Strain", P. 175, case 6)

where

ΔT is any change in torque in the elastic range
$\Delta\theta$ is simultaneous change in angle of twist (radians) in the elastic range

All torque vs. deflection data were plotted to determine ΔT vs. $\Delta\theta$.

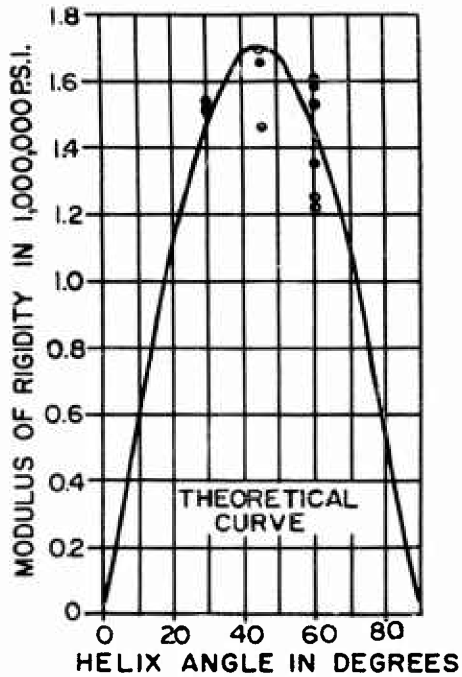


FIGURE 76. MODULUS OF RIGIDITY VS. HELIX ANGLE

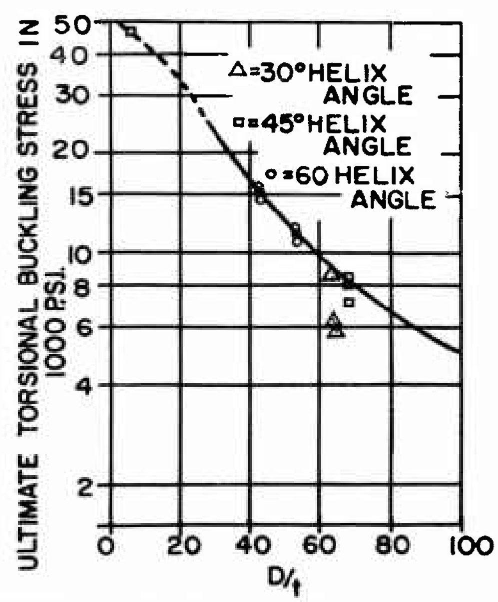


FIGURE 77. TORSIONAL BUCKLING STRENGTH OF HELICALLY WOUND SPIRALLOY TUBES VS. D/t RATIO

TABLE 39

TORSION TEST DATA FOR SPIRALLOY TUBES

HELIX ANGLE	NO. HELICAL LAYERS	DIMENSIONS r_o	r_i	l	TORQUE (ULT.) T	SHEAR STRESS (ULT.) σ_S	D/t	ΔT	$\Delta\theta$	MODULUS OF RIGIDITY G
Degrees		In.	In.	In.	In.-lb.	p.s.i.		In.-lb.	Radians	p.s.i. x 10^{-6}
30	6	2.000	2.065	29	10,550	6,360	64	3,990	.0224	1.50
"	"	"	2.064	28 3/8	10,290	6,310	"	6,350	.0357	1.50
"	"	"	"	28 3/4	13,870	8,510	"	9,490	.0536	1.52
45	"	"	2.059	"	10,500	6,970	68	6,300	.0400	1.46
"	"	"	2.061	29 1/4	12,600	8,090	"	8,400	.0462	1.66
"	"	"	2.059	28 3/4	12,600	8,360	"	7,350	.0402	1.70
60	"	"	2.080	27 3/4	23,100	11,260	51	9,450	.0477	1.29
"	"	"	2.077	27 1/2	22,900	11,600	"	13,650	.0655	1.40
"	"	"	2.081	"	22,680	10,920	"	13,650	.0655	1.31
"	11	"	2.099	29 1/2	39,500	15,750	41	21,000	.0730	1.61
"	"	"	2.100	29 1/8	37,800	14,650		12,580	.0440	1.54
"	"	"	2.099	29 1/4	37,800	15,050	"	16,840	.0588	1.59
45	20	.3	.5	12	8,690	47,980	4	–	–	1.90

INVESTIGATION OF SERVICEABILITY OF SANDWITCH PANELS

The designs of components using sandwich construction are based on the minimum theoretical face thickness consistent with strength requirements. Due to the strength available in reinforced plastics, some of the proposed structures in this report use skins of 0.020 inch thickness. It is recognized that these faces may be thinner than some designers and fabricators consider practical from the standpoint of serviceability.

Several test panels were evaluated to determine the apparent ability of relatively thin reinforced plastic surfaces to withstand damage imposed by normal handling. Such damage may be imparted by tools, walking on surfaces, hand pressure, etc. To reduce the cost of the test program, test panels fabricated for other phases of the program were used after the primary tests had been completed. Seven sandwich panels, each measuring in inches, 15 x 15 x ½, were cut from the original panels as shown in Table 39.

TABLE 39
PANELS FOR SANDWICH SERVICEABILITY TESTS

Number	Face Material	Plies per Face	Core
241-2	181 Polyester	2	Aluminum Honeycomb
241-3	181 Polyester	3	Aluminum Honeycomb
241-4	181 Polyester	5	Aluminum Honeycomb
241-5	181 Epoxy	2	Aluminum Honeycomb
241-7	181 Polyester	3	Fiberglas Honeycomb
241-8	181 Polyester	3	Polyurethane Foam
241-11	181 Polyester	3	Aluminum Multiwave

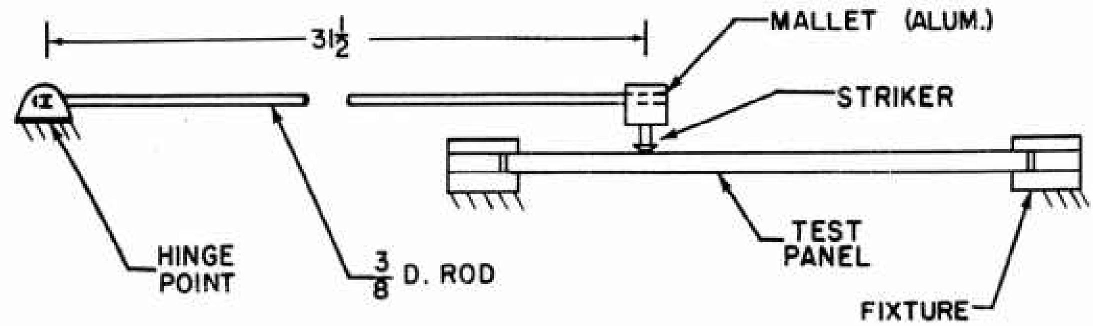
Two aluminum sheet panels of the same size were included for comparison. These were 2024 T-3 alclad with thicknesses of 0.032 and 0.040 inch.

A simple drop test rig was devised for evaluating the panels. It consisted of a pivoted arm with a mallet at the free end to which was attached one of three interchangeable strikers. The test panels were mounted horizontally in a fixture which clamped the edges of the panel, and the mallet was allowed to drop on the panel from various heights. The static weight of the striking head was 0.70 pound. In some cases, an additional weight was attached to increase the total weight to 1.03 pounds. A sketch of the test equipment is presented in Figure 78. Since it is impractical to establish specific standards for acceptability in an evaluation of this nature, the findings must be on a comparative basis. A pointed striker was used to determine the relative resistance to puncture by a sharp object of the various materials tested. The aluminum sheet panels showed superior resistance to puncturing in comparison to the plastic panels with face thicknesses equal to the aluminum. However, the panels with 5-ply polyester facings (No. 241-4) appeared to be equivalent to the .032 aluminum in puncture resistance. The tests indicate that the resistance of the plastic panels is directly related to the number of face plies, with some secondary

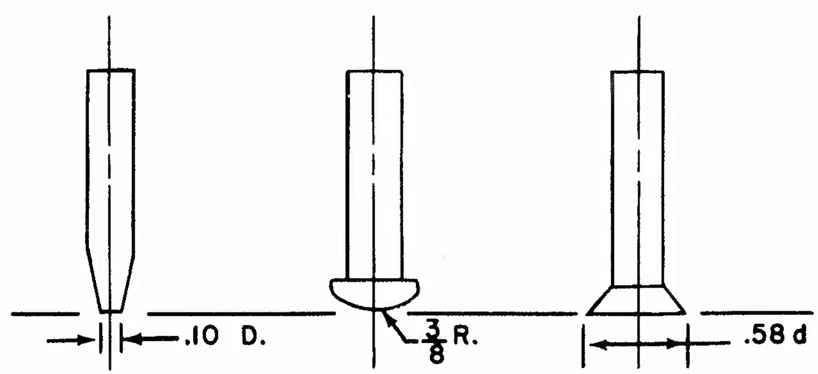
effect from the resiliency of the core. During testing, the two aluminum panels, the 5-ply polyester panel, and the 3-ply polyester panel with urethane foam core (No. 241-8) could not be penetrated with the .70-pound mallet or the 1.03-pound mallet at the maximum drop height of 31½ inches. The panel with the foam core showed evidence of local crushing of the face but did not break through. The two panels with 2-ply facing (No. 241-2 and 241-5) showed a plug the same size as the striker point beginning to shear through when struck with the .70-pound mallet from a height of 24 inches. The other three panels, all having 3-ply polyester facings but different core materials (No. 241-3, 241-7 and 241-11), were punctured with the .70-pound mallet dropped from 31½ inches when the striker hit the panel near the supported edges, but the striker did not quite break through when striking near the center of the panel.

Drop tests with the spherical striker brought out the importance of core resiliency in resisting damage due to impact from a rounded object. The urethane foam panel (No. 241-8) suffered less damage than the other plastic panels, and the area of damage was about the same as the damaged area on the aluminum sheet panels. The panel with the fiberglass honeycomb core was a close second to the foam-filled panel in resistance to damage. Both of these core materials are very resilient as compared to the soft aluminum honeycomb in the other test panels. Tests on the four panels having polyester faces bonded to aluminum cores emphasized the importance of the bond between the face and the core. These test panels, which were known to be defficient in bond strength, were characterized by a distinct light-colored circular area visible on the face after impact. This light area is a result of failure of the bond between the face and the core. The .70-pound mallet dropped 24 inches caused a disk of 1¼- to 1½-inch diameter in the 5-ply facing and a 3/4-inch disk in the 2-ply facing. Evidently, the thicker facing has a greater tendency to pull away from the crushed core. Damage to the 2-ply epoxy panel appeared as slight depressions (max. of 1 inch diameter) with no indications of bond failure except immediately beneath the striker. Compression failure of the core was apparent to some degree even with light loads (6 inch drop) on all panels having the aluminum honeycomb core, but this is considered to have little significance when confined to small areas.

The flat-faced striker imparts a lower unit loading over a greater area than do the other two strikers used in this evaluation. No damage or marks were visible on either the urethane-filled panel or the panel with fiberglass honeycomb core when struck with the 1.03-pound mallet from a height of 31½ inches. Both of the aluminum sheets showed an impression of the striker under similar condition when the impact area was close to the supported edges of the panel. The three panels with polyester facing and plain aluminum honeycomb core behaved under the flat-faced striker much as they did under the spherical striker. The 5-ply facing showed an area of bond failure of about 1/8 inch wide around the perimeter of the striker impression. The aluminum multiwave core showed evidence of higher column strength in the core than the plain aluminum honeycomb. The damage inflicted to this panel by the flat-faced striker was limited to a series of light spots imme-



EQUIPMENT ARRANGEMENT



STRIKER DETAILS

FIGURE 78. SKETCH OF TEST SETUP FOR IMPACT TEST ON SANDWICH PANELS

diately under the striker. The pattern of the spots suggested that they occurred at the intersections of the core ribbons. The 2-ply epoxy panel had a noticeable depression after an 18-inch drop of the .70-pound mallet. The edge of the striker was beginning to cut through the facing after a 24-inch drop. The results of this investigation are summarized in Table 40 by listing the panels tested in descending order of their ability to resist each type of impact load.

TABLE 40
RESISTANCE OF PLASTIC SANDWITCH PANELS
TO IMPACT LOADS

Pointed Object	Rounded Object	Flat Object
.040 Alum.	.040 Alum.	241-8
.032 Alum.	241-8	241-7
241-4	.032 Alum.	.040 Alum.
241-8	241-7	.032 Alum.
241-7	241-5	241-11
241-11	241-2	241-2
241-3	241-11	241-3
241-5	241-3*	241-5
241-2	241-4*	241-4*

*These panels probably would have had better ratings if the quality of bonded joints in the test specimens had been better.

It is concluded from this investigation that the thin-faced plastic sandwich panels are not as effective as aluminum skins of the same thickness ad the facing in resisting penetration by sharp objects. When the impact load is applied by a rounded or flat object, plastic sandwich panels having resilient cores such as urethane foam or fiberglass honeycomb are equal to or better than aluminum sheet having the same thickness as the face ply of the sandwich.

These simple tests are not considered or intended to be used as a guide to determine the serviceability of reinforced plastic sandwich construction for Army aircraft components. They do show, roughly, the comparative resistance of the several materials to specific damage. Although the visable damage to the reinforced plastic sandwich panels appears to be greater than that for metal panels, there is no reason to consider thin-faced sandwich impractical for Army aircraft use.

Small scattered defects of the type inflicted in these tests, resulting in small holes, locally crushed core, indentations, small areas of delaminations, etc., do not affect the strength of the panel appreciably. However, additional care in operation and servicing of the aircraft will probably be necessary to prevent severe damage. Most of the small local

damaged areas can be repaired rather easily. It is concluded that practical structures from a serviceability standpoint can be made of reinforced plastic sandwich. Additional more-extensive research is necessary to determine the effects of damage and the additional care, compared to metal structures, that must be used to prevent damage.

Such a program should also include a study of the sources of damage that a specific structural component might be subjected to: i.e., walking loads; service tools and equipment; stones, etc., thrown by the propeller blast; and other service damage peculiar to the mission and environmental conditions of the aircraft. A test program simulating these and other types of damage would determine practical limits for design.

Effects of Imbedded Conductors and Tubing in Laminates

The objective of these tests was to obtain a preliminary evaluation of the inter-effect between the two materials when bonded together and subjected to loads. A secondary aim was to uncover fabrication problems that may arise in this procedure, including problems due to thermal effects brought about by curing temperatures.

This program consisted of applying varying levels of tension stress to specimens of tubing and wiring imbedded in plastic. Tests were held to the minimum by including only one size of tubing, one size of wire, and two thicknesses of laminate.

Eight tensile specimens were made containing aluminum tube imbedments. Four of these had 8 plies and four had 15 plies. (Ref. Fig 79.) These specimens were pulled in a Baldwin test machine. Test data are given in Table 42. The three specimens which failed, broke at the edge of the end reinforcements. This was as expected since the middle of the specimen was reinforced by the tube. All specimens which had been loaded showed closely spaced lines in the plastic perpendicular to the tubing as shown in Figure 79. The spacing was proportional to the load that had been applied. Minimum spacing was 1/32 to 1/16 inch on the specimens that had failed. When viewed under a microscope, these lines appeared to be cracks in the resin only. They are believed to be indicative of a build-up of stresses adjacent to the rigid aluminum tube.

In order to evaluate the amount of bond between the tube and the plastic, the specimens were sawed in two at the centerline, leaving the tube intact. (Ref. Figure 79.) The plastic could be rotated about the tubing on all the specimens using the force of the hands only. The initial breaking shear stress on the bond was estimated at 100-200 psi.

The failing stress of 17000 to 22000 psi was much lower than had been expected. Two small tension specimens without imbedments were cut from the original specimen as shown in Figure 79. The specimens, when pulled, failed at the same stress as the original. From this, it was concluded that the imbedment was not the cause of the low failing stress.

Six of the eight specimens of laminates with copper wire imbedments were pulled in a Baldwin test machine. Two of these were loaded to failure (ref. Table 42). A slight haze was noted adjacent to the wires when looking through the specimens which had been most heavily loaded. This is thought to be an indication of stress build-up in this area.

A 1/2-inch-long section with a short length of wire protruding was cut from specimen No. 256-1B. When a tension load was applied to the wires,

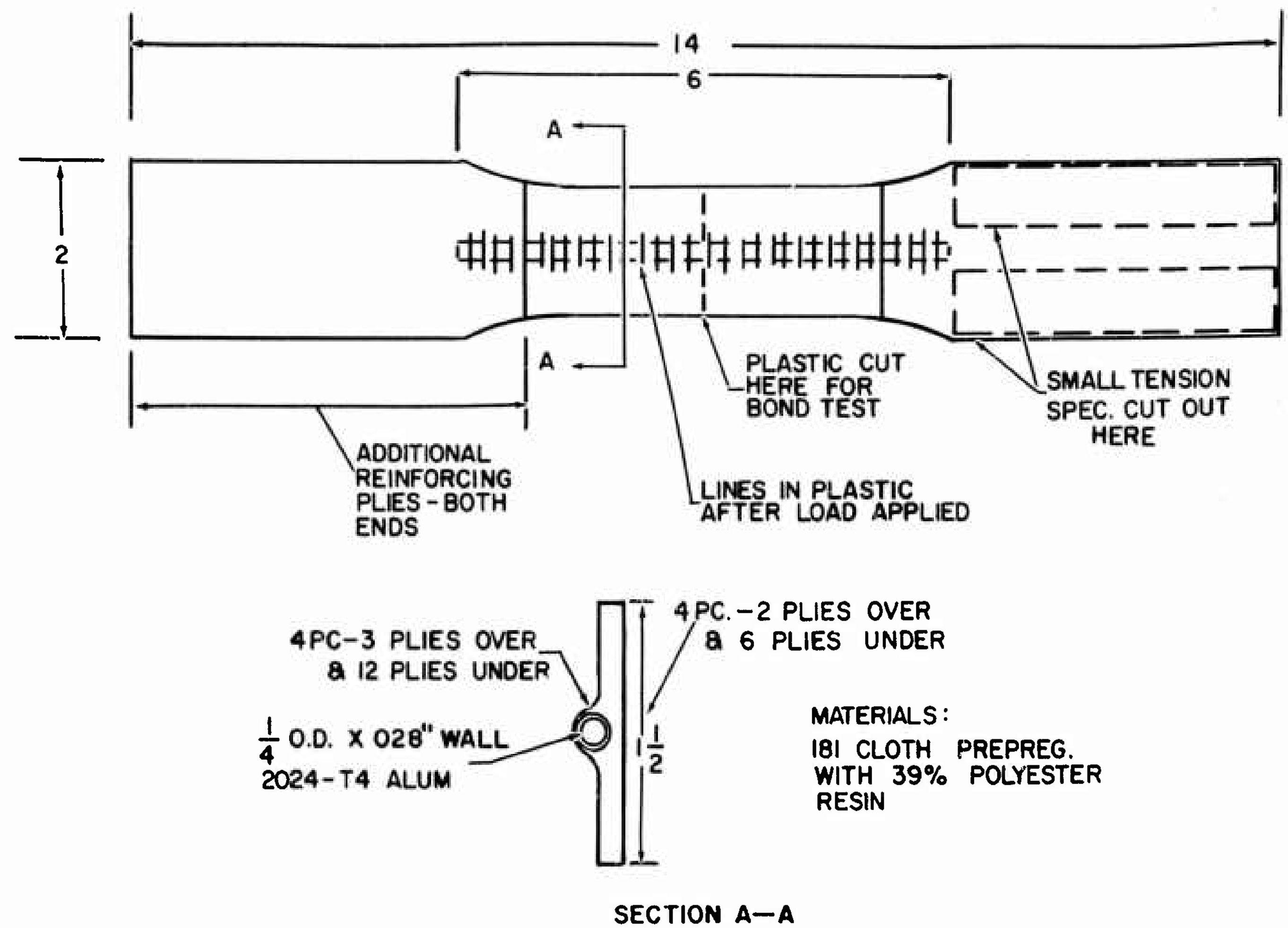


FIGURE 79. TUBE IMBEDMENT TEST SPECIMEN

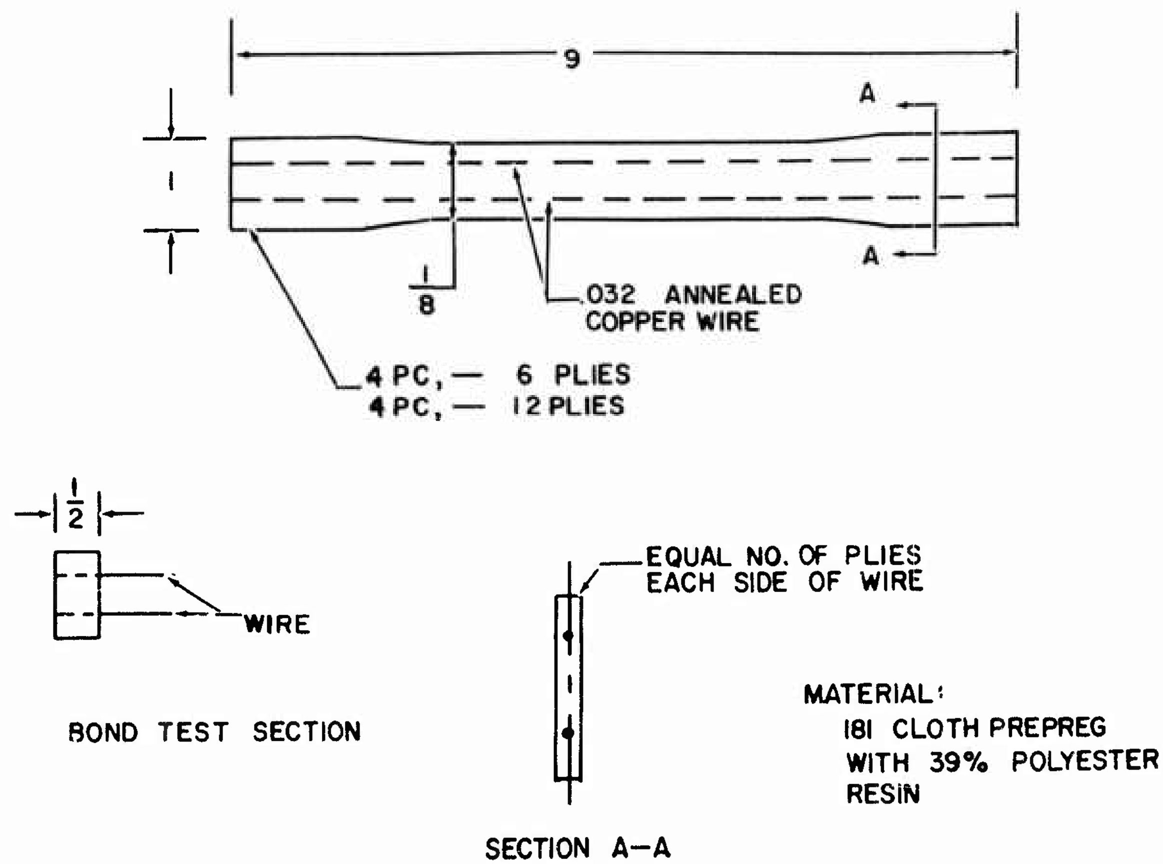


FIGURE 80. WIRE IMBEDMENT TEST SPECIMEN

TABLE 42
TEST DATA - IMBEDDED WIRES AND TUBES

Specimen Number	Number Plies	Imbedment	Load (lb.)	Area (sq.in.)	Stress (p.s.i.)
257-1A	8	Tube	3170*	.141	22500*
257-1B	"	"	3230*	.147	22000*
257-1C	"	"	2200	.148	14800
257-1D	"	"	0	-	0
257-2A	15	Tube	4850*	.285	17000*
257-2B	"	"	4100	.293	14000
257-2C	"	"	2465	.242	10200
257-2D	"	"	0	-	0
256-1A	6	Wire(2)	1800*	.080	22500*
256-1B	"	"	1200	.087	13800
256-1C	"	"	800	.081	9900
256-1D	"	"	0	-	0
256-2A	12	Wire(2)	3700*	.131	28200*
256-2B	"	"	2350	.118	20000
256-2C	"	"	1500	.132	11300
256-2D	"	"	0	-	0

*Failure occurred at this load.

one of the wires came out with a slight pull and the other wire required the application of a tension load of 20 to 30 pounds before bond failure.

The same pull test was conducted on specimen 256-1D, which had not been loaded. The wires broke in this piece at a tension load of 40 to 50 pounds without failing the bond. It is concluded from this test that tension load in specimen 256-1B had appreciably weakened the bond.

Cross-sections of the two specimens were polished and examined under a microscope. Small air-bubble voids were visible on both sides of the wire imbedment. These voids could have a detrimental effect on the strength of the laminate.

Two simulated structural panels containing tubing imbedments were fabricated to evaluate the effect of the thermal expansion properties of the aluminum in plastic. The panels were 8 inches by 30 inches and contained four tubes 1/4 inch in diameter. One panel was made of 6 plies of 181 cloth with polyester resin and the other had 12 plies. After curing, both panels were bowed about 1 1/8 inch out of plane with the tubes on the concave side. The bowing was anticipated because of the greater contraction of the aluminum upon cooling. There was no discernible indication that the bond between the tube and the plastic failed when the specimen cooled.

When fabricating the specimens containing the wires, it was necessary to stretch the wires between pins at each end of the assembly in order to position the wires in the lay-up. In laying-up the specimens containing the tubing, it was found to be advisable to use a narrow strip of fabric to help fill the crevice at the intersection of the tube and the lower layers of fabric.

The tests and examinations conducted were intended to indicate trends and probable results, and were obviously not extensive enough to reach specific conclusions. However, the following generalized statements can be made as a result of this investigation:

1. Metallic imbedments bonded in laminated plastics cause an undesirable build-up of stresses adjacent to the imbedment. This was apparent in the specimens containing the aluminum tubes even though the bond was shown to be very poor.

2. Application of a tension stress as low as 13800 psi can cause a substantial loss of bond between a small copper wire and a plastic laminate.

3. The inclusion of aluminum tubing in a laminated panel will cause appreciable bowing in the panel due to the thermal loads incurred in the curing process.

4. Metallic objects having a round cross-section are difficult to mold into a flat laminate without having voids in the laminate. Special shapes designed to blend in with the lay of the fabric may be a necessity to provide a satisfactory structure.

This study tends to corroborate the opinion that the best approach to the design of laminates with metallic imbedments is to take positive steps to prevent a bond between the metal and the plastic. The problems then resolve into the proper design to prevent weakening of the laminate, design of end fittings, and sealing problems to prevent corrosion due to moisture collecting around the imbedment.

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